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[656 .25 (.492)]

Criticisms of the new speed signalling colour light signal aspects proposed for use on the Netherlands Railways, with replies thereto,(*)

by J.H. VERSTEGEN,

Chief Signal Engineer, Netherlands Railways.

To our great satisfaction we have received from a number of colleagues in different countries remarks, criticisms and expressions of approval concerning our proposals. In this way every aspect of the matter has been thoroughly examined. We are very much obliged to them for these marks of interest and wish to express to all our correspondents our warmest thanks.

Many of these remarks and criticisms are very important and highly interesting. To give a suitable presentation of them we have classified the replies under various headings, indicating at the same time which of the correspondents raised them.

I. *Objections offered to a system of speed signalling aspects developed on a logical basis.*

These objections are principally based on the following points :

a) *Braking distances* are not always uniform.

Reply :

Mr. TRAIN, Mr. TATTERSALL, L.N.E.R., *a*), *b*), *d*).

This is in fact a disadvantage but possesses less importance on the Netherlands Railways seeing that in the Netherlands all trains are fitted with continuous brakes while an increase in the percentage of braking power is contemplated.

(*) See article by Mr. VERSTEGEN, describing the signalling system in question in the *Congress Bulletin* for August 1948, p. 461.

b) *The number of original aspects is too great or the aspects themselves too complicated.*

Reply :

Mr. LASCELLES, *(Railway Gazette)*, Mr. WOODBRIDGE, Mr. TYLER, G.W., *b*), *c*). Mr. ROYLE, L.M.S.R., *b*), *d*). Mr. WALTER, French National Rys., *b*). Mr. Ladislas VARGA, Mr. KULMANN, Hungarian State Rys., *d*).

The number of aspects (10) is the minimum which allows of a 3-speed system functioning.

In fact the following are necessary :

1 red (no height or particular position on the post is laid down for this colour which appears singly [without white]);

2-4 yellow at three positions on the post, in view of its giving warning for *red;*

5-7 green at three positions on the post, and

8-10 the three reduced speed indications :

$$\left.\begin{matrix} green \\ green \\ white \end{matrix}\right\}, \left.\begin{matrix} green \\ white \\ green \end{matrix}\right\} \text{ and } \left.\begin{matrix} white \\ green \\ green \end{matrix}\right\}.$$

To fix our ideas we give here again in four paragraphs the principles of the signal aspects with which trials are now being carried out :

1) In any one signal indication *one* only of the three colours *red, yellow* or *green* can appear. The white lights (lunar white) are complementary and serve solely to enable the driver to determine the position on the post among the three lights showing of any particular coloured light.

2) *Red* appears by itself without any white complementary or « marker » light.

3) *Yellow* can appear in any one of the three positions on the post, but in every case there always appear *one yellow light* and two white ones.

N.B. — *Red* is invariably preceded by *yellow; yellow* is always followed by *red*, that is to say in the 3-aspect system we always have the order *green-yellow-red.* In the 4-aspect system, where there is no difference of height on the post involved (if the distance *yellow-red* falls within braking distance) the sequence is *green, yellow, flashing yellow, red.* (Trials showed that *violet* was not a satisfactory colour.)

In the 5-aspect system, similarly arranged (where this refinement shall prove necessary) the sequence of aspects can become : *green, yellow, flashing yellow* (75 per min.), *flashing yellow* (180 per min.), *red.*

4) *Green* can be shown at any of the three positions on the post, either as *one green* with two white lights or *two greens* with one white light. When *two greens* are showing the uppermost denotes the permissible speed at the signal itself, the other that which the following signal is calling for.

The aspects are logically framed and speak for themselves, their meaning coming out of their very arrangement.

c) It is not practicable to modify the *existing signal aspects* on the lines of those now proposed without bringing about the great inconvenience that temporarily similar aspects will be conveying different meanings.

Reply :

There will be no difficulty in this respect on the Netherlands Railways.

d) The fact that the *direction to be taken by the train is not shown at junctions* is represented as a disadvantage in previously existing systems.

Reply :

In our opinion speed is the dominant factor and direction secondary. It would always be possible to add some sign of direction to the signal aspect (for example in the form of an arrow (junction indicator) or « marker » at those places where it is considered to be absolutely necessary, as at junctions between stations with the *same* speed permissible on both branches.

II. *Objections to the form of the signal itself.*

a) Some consider *the signal showing 3 lights to be too complicated.*

Reply :

Mr. WALTER, French National Rys., *a*), *b*), *c*).
Mr. Gunnar KAVLI, Norwegian Rys., *a*).

In a three-speed system, it is only by means of aspects showing 3 lights that such indications, conveying their meaning inherently in themselves are possible. For a very long time such three-light aspects have been in service in America where, it is true, they are used with another purpose and meaning and even with more than ten aspects, which however do not obviously convey their meaning from

their appearance. In addition it must be pointed out that by its very nature a signal indication is not complete unless it does show three lights. (Two light aspects are seen on the Netherlands Railways only in the case of certain dwarf signals.) In the case of aspects which are not burning or are imperfectly showing, special instructions are given, under which no incomplete aspect can at the worst authorise a speed higher than that which it would allow of if properly exhibited. It is only the failure of a *yellow* light to flash that can constitute a danger since the light then would become a steady one, but this is provided for by so arranging matters that if the flashing fails the lights shall go out altogether.

b) The application of a *white light in the signal aspects* brings about a *reduction in visibility.*

Reply :

In our arrangement of fittings and using the « lunar white » light in the « searchlight » type of signal unit, we have not found any drawback to exist in this system. It must be borne in mind that :

1) *Red* is very clearly visible and can be perceived at any distance desired;

2) *Yellow* is equally easy to distinguish. Shown at the top or at the bottom it is observable also at a great distance. Showing from the middle unit it cannot be seen so far, although from the fact that *yellow* is seen to be neither at the top or the bottom it is possible to conclude that it is being shown from the centre unit, it is nevertheless of interest to endeavour to ascertain at what distance the driver can be sure that this is the case. Strictly speaking this should only take place at the signal itself, seeing that it has to be preceded by either : *green* showing from the middle unit (in which case it is only necessary to take action after the *yellow* signal); *green* at the top and in the middle, in which case the speed at the *yellow* should have been brought down to medium (indicated by the middle *green* in the preceding signal). (In the case of fog, action is only taken after passing such preceding signal.) Taking into account the fact that a signal aspect may become more favourable after a train has passed the preceding one (which itself was calling for a reduction of speed) it is desirable in order to avoid an unnecessary reduction, in clear weather, to be able to see

clearly the middle *yellow* light at the point where this reduction in speed should commence. In the case under consideration this is therefore somewhere before applying the brakes to reduce from high to medium speed. The signal aspect fulfils this requirement more than is actually necessary.

3) If we consider in like manner the aspects formed with the aid of the *green* light, we find that such *green* light is equally distinct at a great distance in either the top or bottom unit, but less so when showing from the middle one. Here again we have to be satisfied with the requirement above mentioned and this is quite fully met.

The most unfavourable condition that we can think of would be that in which a driver perceived insufficiently clearly either *yellow* or *green* showing from the bottom unit while the speed was still high (the preceding signal showing *green* at the top and at the bottom). He ought to be able in that case to perceive the signal aspect distinctly at the point where braking distance from high to low speed begins. This also is satisfactorily complied with.

c) The « searchlight » type of signal is *less reliable in operation* than the type having *no internal spectacle mechanism.*

Reply :

Our many years experience of this type of signal and practice in many other countries (America for example) have proved that these signals are not less reliable in working than other kinds while they offer in addition considerable advantages.

d) The number of lights showing at the same time could be *reduced* by using *flashing lights.*

Reply :

Mr. HARD, Swedish State Rys., *d*).

That is so but it would be detrimental to the clearness of character of the signal aspects besides which it is found in practice that the running staff raises objections to the general application of flashing lights and for this reason the use of this more complicated arrangement has to be kept to a minimum.

Finally it is appropriate to point out here that the speeds authorised by the signal aspects are different for passenger and freight trains :

High :	125 km. (77 miles)/h., later 135 km. (83 miles)/h., for passenger trains; 75 km. (46 miles)/h., later 90 km. (56 miles)/h., for freight trains.
Medium :	90 km. (56 miles)/h. for passenger trains and 50 km. (31 miles)/h. for freight trains.
Low :	45 km. (28 miles)/h. for passenger trains and 30 km. (18 miles)/h. for freight trains.

In conclusion we express once more our cordial thanks for the interest that has been shown in this question, and the hope that it will have the practical result of inducing colleagues, who may be sure of receiving a hearty welcome from us, to come and see the system in operation.

[621 .135 (01, 625 .14 (01 & 625 .22]

Superelevation and maximum speeds as functions of the radius of curves and of gradients,

(*Continued* *)

by DEMOSTH. PROTOPAPADAKIS,

Civil Engineer,

Professor of (Railway) Engineering, National Polytechnic School, Athens.

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(*) See *Bulletin of the International Railway Congress Association* for May 1948, p. 286.

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PART TWO.

Chapter 1.

A. FOREWORD.

1. Introduction.

In the working time tables train crews are given average speeds between consecutive stops, as well as the time allowed for the distance.

The speed thus prescribed should not exceed the smaller of the two speeds $V_\rho = f(\rho)$ or $V_i = \varphi(i)$, calculated according to the characteristics of the section $\rho_{min.}$ and i_{max}. Curves of radius $\rho' > \rho_{min.}$ would be suitable for a speed $V_{\rho min.} = f(\rho)$ if $V_i > V_\rho$; advantage will be taken of this to give them a cant of $d' = d\rho$; ρ' less than that of d, provided for curves of radius ρ. On gradients $i' < i$, the speed $V_i = \varphi(i)$ will be attained if $V_i < V_\rho$.

In the article published in the *International Railway Congress Bulletin* dated December, 1933, mention was made of the great diversity between the formulae $V_\rho = f(\rho)$ and $V_i = \varphi(i)$ used by the various Administrations. From this it will be seen that the Administrations are not agreed on the value of average running speed or on the max. speed V_{max} most suitable for each speed range, comprising frequent curves of minimum radius $\rho_{min.}$ or max. gradient i_{max}.

A comparative study of the various formulae in use has led to the establishment of Table No. I and the corresponding Graph No. II, in this article as well as Table No. III and Graph No. IV. A brief scrutiny of these leaves no doubt about the differences of opinion; it will be seen, in fact, that they increase with the values of V, ρ and i.

In this article, we have considered it advantageous to :

1. give some explanation of the general form of the formulae $V_\rho = f(\rho)$, $V_i = \varphi(i)$ and $\rho = \varphi_1(i)$ and their actual interpretation;

2. give some concise information on the formulae included;

3. indicate in Table No. IV the reactions resulting from the speeds V_ρ or V_i used on other graded or level curves and their effects from the point of view of 1) smooth running and 2) wear — greater or smaller — on rails and wheel tyres;

4. oppose frequent and, more particularly, large variations in plan or profile of the rail alignment;

5. stress the considerable increase in resistance, particularly the opposing forces in still air, with increasingly high speed;

6. stress the necessity for stronger opposition to the tendency to increase speeds beyond 90-100 km./h. or 105 km./h. (56-62 or 65 m.p.h.).

We may note finally that the present article is considered complementary to that of the December, 1933, *Bulletin*, mentioned above; reference is made to this to avoid tedious repetitions and to emphasise that the themes of the two articles are similar, i.e. the difference of opinion on the limit of average running speed between consecutive stops and the effects thereof.

2. Symbols and formulae used.

The following are used :

$V_{min.}$ — the speed of the slowest trains over the section which has $\rho_{min.}$ and $i_{max.}$ as characteristic elements of the alignment, in km./h.;

V — the max. running speed over the section, in km./h.;

V_0 — the max. speed authorised by the Administration (120 km./h. in France [75 m.p.h.]);

V_1 — the speed used in the theoretical formula for calculating superelevation, in km./h. (neutral speed);

v_a — the speed at which maximum efficiency of the motors is recorded (in m./sec.);

v — the speed corresponding to V ($v = V : 3.6$) in m./sec.;

ρ' — all radii in the section greater than the radius ρ or, exceptionally, smaller than ρ, in metres;

i_n — gradients in excess of which it is necessary to brake, in mm./m.;

$e = 1\,500$ mm. The distance between the rail axes for normal gauge track;

h — the superelevation calculated for V_1 on curves of radius ρ, in mm.;

h_0 — the maximum superelevation (= 150 mm. for this article) in mm.;

h' — all other superelevation in curves of radius ρ of the section ($h' = h\rho : \rho'$) in mm.;

d, d_0 and d' — the cant corresponding to the superelevations h, h_0 and h' ($d = \frac{h}{e}$, $d_0 = \frac{h_0}{e}$, $d' = \frac{d\rho}{\rho'}$) in mm./m.;

a_e and a_i, p_e and p_i — the accelerations and specific thrusts applied to the exterior and the interior rails of curves of radius ρ, also on the vehicles, in m/sec² and kgr./kgr. of the weight of the vehicles, given by the formulae :

$$a_e = \frac{V^2}{12.96\rho} - g\frac{h}{e}$$

$$p_e = a_e : g = 0.102 a_e$$

$$a_i = g\frac{h}{e} - \frac{V^2}{12.96\rho}$$

$$p_i = a_i : g = 0.102 a_i ;$$

S — the maximum transverse section of a normal gauge train, taken here as 11 m²;

R_a — $0.006V^2 \times S = 0.066V^2$ the resistant effort in still air to the movement of a normal gauge train, in kgr.;

T_a — $0.0002444...V^3$ the work corresponding to the resistance R_a in units of 75 kgr./sec.;

m' — the *mass of inertia* of a train, including motor ($m' = 1.08$ to 1.25 of the mass $m = P : g$);

E_a — the effort corresponding to the speed v_a, in kgr.;

$w = a + bv^2$ or $a + bv + cv^2$ — the specific resistance of a train running at a speed v, in kgr.;

$T_w = av + bv^3$ or $av + bv^2 + cv^3$ — the resistant work corresponding to the speed v, in kgrm.;

r' — the total time losses for stopping and starting, in minutes;

t_0 — the braking time, in seconds;

s_0 — the stopping distance, in metres;

$E_n = \frac{1}{2}m'v^2$ — the kinetic energy of the train;

e_n — (0.0042473 up to 0.004326)V^2 per kgr. of train (*);

p — braked weight, in tons;

P — total weight of train, in tons;

$\lambda = p : P$, coefficient of braking;

r — brake resistance, in kgr./t.

3. a) Formulae $V = f(\rho)$ and their rational application.

The formulae used are mostly the general form :

$$V = \alpha\sqrt{\rho - \beta} \quad . \quad . \quad . \quad . \quad (a)$$

in which α is a numerical coefficient which varies from 2.85 to 4.85 and even to 5.00 *for motored sets;* β is a positive, zero or negative number, according to whether the speed V is smaller, equal to, or larger than that which would be given by the formula $V = \alpha\sqrt{\rho}$; β itself can be a function of ρ; it is equal to

$$0.00015803\rho^2 + 64.6834$$

in the type

$$V = \sqrt{-(0.00015803\,\rho^2 + 64.6834)}$$

which gives, very approximately, the maximum speeds V as a function of ρ for the German railways. The lower limit of ρ would be that at which the motor could enter into, but not advance along, the curve. We believe that, in the case of normal gauge track, radius ρ lower than those in the first column of Table No. I should not be used.

The upper limit of ρ, for the application of formula (*a*) will be given by the condition $\alpha\sqrt{\rho - \beta} \leq V_0$, that is to say, by the condition $\rho_1 \leq \frac{V_0^2}{\alpha^2} + \beta$.

(*) For masses of inertia $m' = 1.08$ and 1.50.

For radii larger than ρ_1, it is always taken that $V = V_0$. Thus, for *high, average and low maximum speeds*, i.e. by the application of formulae 13, 12 and 10 of Table No. I, the maximum speed of 120 km./h. (75 m.p.h.) is reached with radii of 711, 1 024 and 1 600 m., beyond which it is always taken that $V = V_0 =$ 120 km./h. without any calculation.

We may add that there exist other types of formulae; this is the case, for example, on the French PLM., which states

$$V = \begin{cases} 25 + 0.16\rho \text{ for } \rho \leqslant 333 \text{ m.} \\ 45 + 0.10\rho \text{ for } \rho \geqslant 333 \text{ m.} \end{cases}$$

This is also the case with the formula :

$$V = \sqrt{12.96(a_e + a_i)\,\rho + V^2_{min}},$$

set out in the December, 1933, *International Railway Congress Bulletin* article, and which shews that V is dependent not only on the value of ρ, but also on the accelerations a_e, a_i and quite as much on the value of V_{min}.

b) Rational use of the formulae $V_\rho = f(\rho)$ and $V_i = \varphi(i)$.

The application of these formulae for *each* radius ρ or for *each* gradient i of a section is impossible for several reasons, three of which are essential :

1. The train crew cannot know at any given time or point on the line the values of ρ and i, in order to observe the relative speeds.

2. Even if they knew these values, they could not maintain the relative speeds, i.e. constantly and immediately vary the speed v of the train, owing to *its large kinetic energy* $E_n = 0.50\ m'v^2$ and *its low mass power* $P_{m'} = 0.50v^2$, having regard to the low inert mass m'.

3. Each motive unit on rail produces its maximum efficiency $E_a v_a$ when running at a speed v_a, or at a speed v'_a which is little more or less than v_a. Its efficiency will thus be lower the further the speed is from v_a, above or below v_a.

These three reasons are also valid for *automobiles* but to a less degree because of the low mass in proportion to the power of the motor; it has *a low kinetic energy* and *high mass power* and its running speed can consequently be more easily varied.

To determine average running speed therefore, we do not apply the speed provided at each moment of application of the formula $V_\rho = f(\rho)$ or $V_i = \varphi(i)$, but the speed resulting from *a single radius* ρ or a single gradient i previously defined as the *most frequent minimum radius* or *most frequent maximum gradient on a speed range.* It is desirable from a point of view of the efficiency $E_a v_a$ of the train motor if the maximum average speed thus calculated is not very different from $3.6\ v_a$ (having regard to the kinetic equation $v = V : 3.6$).

The foregoing facts lead to the rational use of the formulae giving the maximum speeds as a function of the most frequent minimum radius ρ or maximum gradient i over the section being considered. The principles to be followed are : 1) to divide the line into *sections for speeds*, without large or frequent variations in curvature or gradient in the same section; 2) to establish the characteristics ρ and i for each section; 3) to balance ρ and i so that the speed V, calculated as a function of $V_\rho = f(\rho)$ or of $V_i = \varphi(i)$ is the same; if this is not so it will be necessary to take the lower of the two speeds V_ρ or V_i, without profiting from the higher. Taking an example from the German Railways, if the speed section

has as characteristics $\rho = 500$ m. and $i = 12$ mm. the average running speed for the two characteristics would be 85 km./h. (52 m.p.h.), whilst on the same section with characteristics $\rho = 400$ m. and $i = 12$ mm. the speeds would be 75 km./h. (46 m.p.h.) for radius and 85 km./h. for gradient, and the smaller speed would have to be taken, i.e. a loss of 10 km./h. (6 m.p.h.).

B. Formulae $V = f(\rho)$ used and formulae proposed.

Table No. I and Graph. No. II.

1. *Formula No. 1.*

$$V = 4.82\sqrt{\rho - 27}. \quad . \quad . \quad . \quad (1)$$

This is quoted in « Costruzioni stradali e ferroviarie », p. 574, by Vittorio BAGI, for the Italian system. It also gives the speeds shewn in table 28*a*, p. 265, of that publication « Costruzioni stradali e ferroviarie », by L. STABILINI, for the same system.

It also gives, with a 5 km./h. (3 m.p.h.) excess, the maximum speeds V adopted by the French PLM (see formula No. 2).

The speeds given by the formula are the highest in Table No. I; they differ little from those given by the formula No. 13, $V = 4.5\sqrt{\rho}$ of the same table, the formula which is proposed *for highest maximum speeds* in place of formula No. 1.

The speeds given in the work by L. STABILINI mentioned above, are based on a superelevation of $h = 150$ mm. and go up to 150 km./h. for $\rho = 1\,000$ m. Allowing $V_0 = 120$ km./h., we see that beyond $\rho = 650$ m. formula No. 1 gives speeds greater than V_0; for $\rho > 650$ m. a maximum speed of 120 km./h. is therefore taken without calculation.

Curve 13 of Graph No. II represents the speeds given by formula No. 13 of Table No. I, as well as those given by formula No. 1.

2. *Formula No. 2.*

$$V = 4.3\sqrt{\rho}. \quad . \quad . \quad . \quad (2)$$

of the French PLM, which uses for its maximum speeds V the two formulae :

$$V = \begin{cases} 25 + 0.16\rho & \text{for } \rho \leqslant 333 \text{ m.} \\ 45 + 0.10\rho & \text{for } \rho \geqslant 333 \text{ m.} \end{cases}$$

with a maximum superelevation of 160 mm.

The superelevations h' are calculated by the formula $h' = 160\ \rho : \rho'$.

Under No. 2 the representative lines of the two latter formulae have been traced. They are quoted in the « Cours lithographié de Chemins de Fer », by M. MARTINET of the « Ecole Nationale des Ponts et Chaussées de Paris », Session 1932-1933, p. 15, Part 1. They have been replaced by the single formula No. 2 which gives almost the same speeds without any angularity in the curve; moreover, it is more easily compared with the other formulae.

For the angular point $V = 78.33$ km. per hour has been taken according to the PLM formula and $V = 78.467$ km./h. in accordance with the substituted formula No. 2.

3. *Formula No. 3.*

$$V = 4.64\sqrt{\rho} - 130 \quad . \quad . \quad (3)$$

of the French Est, quoted as

$$V = \sqrt{22\rho - 3\,000}$$

in the « Cours lithographié des Chemins de Fer », by M. DESCUBES of the « Ecole Nationale des Ponts et Chaussées de Paris », Session 1921, p. 6, Part 1, and p. 3

of Appendix 2. It was drawn up by M. DESCUBES to give speeds of 60, 90 and 120 km./h. on the speed sections aligned with minimum curvatures of 300, 500 and 800 m., with a superelevation limit of 150 mm.

The superelevations y are calculated by the formula $h = 12\frac{V_1^2}{\rho}$ or $V_1 = \frac{2}{3}V + 20$ for $V \geqslant 60$ km./h.; for lower speeds $V_1 = V$ is taken. For curves $\rho \geqslant 800$ m. it is always taken that $V = V_0 = 120$ km./h., the maximum speed allowed on the French Railways. The formula was proposed for $\rho \geqslant 200$ m.

It may be noted that for values of ρ between 300 and 800 m., and for ease of calculation, the formula above may be replaced by :

$$V = \rho(2\,300 - \rho)10^{-4}$$

which gives, very exactly, the speeds of 60, 90 and 120 km./h. for radii of 300, 500 and 800 m. This formula gives higher speeds than DESCUBES' for $\rho < 300$ m. and smaller speeds for $\rho > 800$ m.

The formula in question has been set out in the form No. 3 $V = 4.64\sqrt{\rho} - 130$ (3) in order to estimate better its position in relation to the other formulae of Table I. Curve No. 3 in Graph II represents this formula.

4. *Formula No. 4.*

$$V = 4\sqrt{\rho} \quad . \quad . \quad . \quad (4)$$

by HOOD, proposed for the London and South Western Railway.

This formula, along with Nos. 5, 7 and 10, is quoted in the « Enzyklopädie des Eisenbahnwesens », Vol. 8, p. 338, 1917. Curve No. 4 in Graph II shews it.

5. *Formula No. 5.*

$$V = 4.25\sqrt{\rho - 70} \quad . \quad . \quad (5)$$

of the Austrian Railways.

The speeds given differ very little from those given by the English formula; it has, therefore, not been included in Graph II.

6. *Formula No. 6.*

$$V = 4\sqrt{\rho - 59} \quad . \quad . \quad . \quad (6)$$

of the German Railways.

This formula is drawn up so that we may get as nearly as possible the speeds V of the German Railways, noted in Vol. 3, p. 885, of the 26th. edition (1934) of the « Hütte » manual, and shewn, with the superelevation h on p. 891.

The « Henschel Locomotive Engineers Manual », 1935, p. 23, states that « the German Railways intend to increase speeds to V = 100, 120, 140, 160 and 200 km./h. for curves of ρ = 400, 600, 800, 1 000 and 1 600 m. ».

If these speeds are realised, intermediate speeds can be calculated by formulae similar to that of the French PLM, that is :

$$V = \begin{cases} 60 + 0.10\rho \text{ for } \rho \\ \text{between 400 and 1 000 m.} \\ 93 + 0.067\rho \text{ for } \rho \\ \text{between 1 000 and 1 600 m.} \end{cases}$$

Maximum superelevation on the German Railways is fixed at 120 mm. (see the « Hütte » Manual, p. 891, vol. as above). It is calculated by the formula :

$$h = \frac{8V^2}{\rho} + \varepsilon.30$$

by taking :

$\varepsilon = 0$ for *smooth running;*

TABL

Speed

Order.	Formulae : V =	Characteristi 150	200	250	300	350	400	450	500	550
1.	$4.82\sqrt{\rho - 27}$	53	63	72	80	87	93	99	105	110
2.	$4.3\sqrt{\rho}$	53	61	68	74	80	86	91	96	101
3.	$4.64\sqrt{\rho - 130}$	...	39	51	60	69	76	83	89	95
4.	$4\sqrt{\rho}$	49	57	63	69	75	80	85	89	94
5.	$4.25\sqrt{\rho - 70}$	38	48	57	64	71	77	83	88	93
6.	$4\sqrt{\rho - 54}$	38	47	55	62	68	74	79	84	89
7.	$4\sqrt{\rho - 50}$	40	49	57	63	69	75	80	85	89
8.	$3.6\sqrt{\rho}$	44	51	57	63	67	72	76	80	84
9.	$3.1\sqrt{\rho + 140}$	53	57	61	65	69	72	75	78	81
10[1].	$3\sqrt{\rho}$	*37*	*42*	*47*	*52*	*56*	*60*	*64*	*67*	*70*
11.	*Averages :*	45	51	59	65	71	76	81	86	91
12[2].	$3.75\sqrt{\rho}$	*46*	*52*	*59*	*65*	*70*	*75*	*80*	*84*	*88*
13[3].	$4.5\sqrt{\rho}$	*55*	*64*	*71*	78	*84*	*90*	*95*	*101*	*106*
14.	$3\varepsilon\sqrt{\rho}$									

ε = 1.00, 1.25 or 1.50 for the *low*

$\varepsilon = +1$ for the *greatest superelevation;*

$\varepsilon = -1$ for *the smallest superelevation.*

all three having regard to the same speed V, calculated by the formula :

$$V = f(\rho) \approx 3.85\sqrt{\rho}$$

Curve No. 6 of Graph II shews the speeds given by formula No. 6.

It has a regular appearance without any of the polygonal variations which correspond to the speeds set out in the « Hütte » manual.

7. *Formula No. 7.*

$$V = 4\sqrt{\rho - 50} \quad . \quad . \quad . \quad . \quad (7)$$

known as the RUPPEL formula.

This formula gives, as does the formula No. 12 proposed by us as for *medium average speeds,* almost exactly the *average speeds* shewn in the 11th line of Table No. I. It has been preferred to formula No. 12 on account of the correlation of the latter to formulas Nos. 10 and 13 which we propose *for low and high maximum speeds.*

This formula is not included in Graph No. II because of its similarity to formula No. 12 which is included.

o. I.

$= f(\rho)$.

adii ρ :

600	650	700	750	800	850	900	950	1 000	1 100	1 200
115	120*	125	130	134	138	142	146	150	158	165
105	110	114	118	122*	125	129	133	136	143	149
101	106	111	116	120*	125	129	133	137	145	152
98	102	106	110	113	117	120*	123	126	133	139
98	102	107	111	115	119	122*	126	130	136	143
93	97	101	105	109	112	116	119*	123	129	135
94	98	102	106	110	113	117	120*	123	130	136
88	92	95	99	102	105	108	111	114	119*	125
84	87	90	92	95	98	100	102	105	109	114
73	*76*	*79*	*82*	*85*	*87*	*90*	*92*	*95*	*99*	*104*
95	99	103	107	110	114	117	120*	125	130	136
92	*96*	*99*	*103*	*106*	*109*	*112*	*116*	*119**	*124*	*130*
110	*115*	*119**	*123*	*127*	*131*	*135*	*139*	*142*	*149*	*156*

average[2] or *high*[3] *maximum speeds.*

It will be noted that the speeds given by the Ruppel formula are those given by the Hood formula for radii ρ from 50 m.

8. *Formula No. 8.*

$$V = 3.6\sqrt{\rho} \quad . \quad . \quad . \quad . \quad (8)$$

This formula has been proposed by several authors and arises from the theoretical incline formula $h : e = V^2 : 127.14\rho$ which should be constant for any gauge of track and equal to 0.10; this would give

$$V = \sqrt{127.14\rho} = 3.565\sqrt{\rho} \approx 3.6\sqrt{\rho}$$

This gives 96 % of the *maximum average speeds;* divided by 3.6 it gives the train speed in m./sec... $v = \sqrt{\rho}$.

It is shewn in Graph II by curve No. 8.

9. *Formula No. 9.*

$$V = 3.1\sqrt{\rho + 140} \quad . \quad . \quad (9)$$

This formula was set up in our article in the *International Railway Congress Bulletin* of December 1933, for a super-elevation of $h = 70 + \frac{14\,000}{\rho}$.

These two formulae were drawn up for the particular values $a_e = 0.30$, $a_i =$

0.45 and $V_{min.} = 35$ km./h. so as to *give smooth running and low wear of rails and tyres.*

By taking the extreme limits $a_e = 0.40$, $a_i = 0.60$ and $V_{min.} = 25$ km./h., the formulae :

$$V = 3.6\sqrt{\rho + 50} \text{ and } h = 92 + \frac{7\,375}{\rho}$$

are obtained, the first of which obviously gives the *average max. speeds;* they are those of formula No. 8, increasing ρ by 50 m.

If we take the values $a_e = 0.30$ and $a_i = 0.60$ as maximum values with $V_{min.} = 25$ km./h., the formulae :

$$V_{max.} = 3.4\sqrt{\rho + 55} \quad . \quad . \quad (a)$$

$$\text{and } h = 90.51 + \frac{7\,500}{\rho} \quad . \quad . \quad . \quad . \quad (b)$$

are easily obtained.

These two formulae are preferred to those above and are also preferred to those set up in the Dec., 1933, *Bulletin.*

Curve No. 9 of Graph No. II represents formula No. 9, and its position, relative to curves Nos. 10, 12 and 13, is readily seen, being — beyond 300 m. — between *the low and medium average speeds.*

10. *Formula No. 10.*

$$V = 3\sqrt{\rho} \quad . \quad . \quad . \quad . \quad . \quad (10)$$

The provision of this formula has been noted under formula No. 4. It gives the *smallest maximum speeds* shewn on line No. 10 of Table No. I. Increasing its numerical coefficient 3 by 25 % and 50 %, we obtain formulae Nos. 12 and 13 giving *medium and high average speeds.*

Curve No. 10 of Graph No. II shews the speeds of this formula.

In the Hoepli Manual « Trazione a vapore », G. OTTONE, pp. 157-158, Milan, 1905, are shewn the formulae $V = 2.846\sqrt{\rho}$ and $V = 2.846\sqrt{\rho} + 5$, the results of which differ little from those of formula No. 10.

11. *Averages.*

Under this heading have been shewn the arithmetical averages *of each column* of the first ten lines of Table No. I.

We have calculated these averages to shew that they differ little from *the average maximum speeds given* by formula No. 12, which we propose for the calculation of the latter speeds.

12. *Formula No. 12.*

$$V = 3.75\sqrt{\rho} \quad . \quad . \quad . \quad . \quad (12)$$

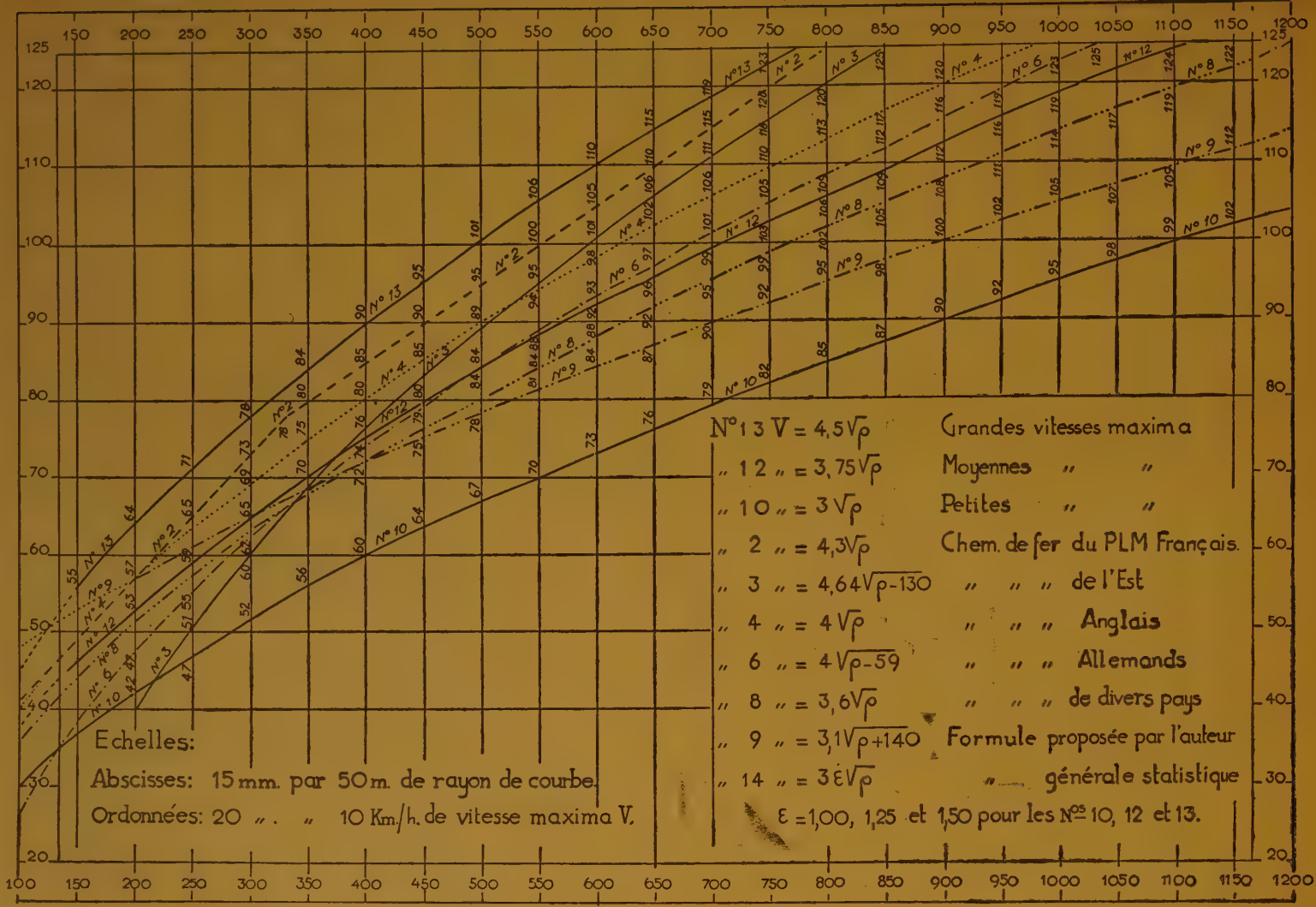
This formula has been drawn up to give, as nearly as possible, the *averages* of the 11th. line of the Table No. I. It is obtained by increasing by 25 % the numerical coefficient 3 of formula No. 10; in Graph No. II, it is shewn by curve No. 12.

13. *Formula No. 13.*

$$V = 4.5\sqrt{\rho} \quad . \quad . \quad . \quad . \quad (13)$$

It has already been stated under formula No. 1, why formula No. 13 has been introduced for *highest maximum speeds.* It can be readily seen that the speeds given by it differ very little from those of formula No. 1.

The formula has been shewn in Graph No. II by curve No. 13. It gives an increased speed of 12.5 % by comparison with the English formula by HOOD.


N° 13 V = 4,5√ρ
„ 12 „ = 3,75√ρ
„ 10 „ = 3√ρ
„ 2 „ = 4,3√ρ
„ 3 „ = 4,64√ρ-130
„ 4 „ = 4√ρ
„ 6 „ = 4√ρ-59
„ 8 „ = 3,6√ρ
„ 9 „ = 3,1√ρ+140
„ 14 „ = 3ε√ρ
ε =1,00, 1,25 et 1,50 pour les N°s 10, 12 et 13.
Grandes vitesses maxima
Moyennes „ „
Petites „ „
Chem. de fer du PLM Français.
„ „ „ de l'Est
„ „ „ Anglais
„ „ „ Allemands
„ „ „ de divers pays
Formule proposée par l'auteur
„ générale statistique
Echelles:
Abscisses: 15 mm. par 50 m. de rayon de courbe.
Ordonnées: 20 „ . „ 10 Km/h. de vitesse maxima V.

N.B. — Scale : *Abscissae :* read 9 mm. instead of 15 mm. — *Ordinates :* read 12 mm. instead of 20 mm.

Grandes vitesses maxima = High maximum speeds.
Moyennes vitesses maxima = Medium maximum speeds.
Petites vitesses maxima = Low maximum speeds.
Chem. de fer du PLM Français = French PLM.
» » » de l'Est = French Est.

Explanation of French terms:

Chem. de fer Anglais = English railways.
» » » Allemands = German railways.
» » » de divers pays = Railways of various other countries.
Formule proposée par l'auteur = Author's proposed formula.
» générale statistique = General statistical formula.
E = 1,00, 1,25 et 1,50 pour les N°s 10, 12 et 13 = 1.00, 1.25 and 1.50 for the Nos. 10, 12 and 13.

Statistical summary of Table No. I.

14. General formula No. 14.

$$V = 3\varepsilon\sqrt{\rho} \quad . \quad . \quad . \quad . \quad (14)$$

From all the foregoing, we can extract the following statistical result :

On normal gauge speed sections, maximum speeds given by the general formula No. 14 are used, introducing :

1.00 for the *low maximum speeds;*

1.25 for the *medium maximum speeds;*

1.50 for the *high maximum speeds.*

on condition that the maximum characteristic gradient allows a speed of $V_i = \varphi(i)$ equal to or larger than $V_\rho = f(\rho)$.

C. Formulae $V_i = \varphi(i), \rho = \varphi_1(i)$ and their rational use.

There is a large number of these formulae; they are old and are no longer applied in view of the great evolution, particularly as regards the braking of rolling stock and the continuing tendency towards high speeds.

We have summarised in Tables II and III and the corresponding Graph No. III, the figures and formulae proposed to facilitate their comparison. Thus :

a) Formulae $V_i = \varphi(i)$.

1. *On line 1* of Table III (*) we have shewn the speeds V_i proposed by Max Edler VON LEBER (see *International Railway Congress Bulletin*, August 1892) for the Austrian system. They were included in the diagram on p. 158 of the manual « Trazione a vapore », by G. OTTONE. Curve No. 1 of Graph No. III shews them.

2. *On line 2* are shewn the speeds V_i proposed in the article by Prof. E. VICAIRE *(Annales des Ponts et Chaussées*, March, 1880, p. 234). mentioned as

Grades :	0 ÷ 5	5 ÷ 10	10 ÷ 15	15 ÷ 20	20 ÷ 25	25 ÷ 30	mm./m.
Speeds :	100	85	70	55	40	25	km./h.

and calculated in the table by the linear formula $V_i = 115 - 3i$, which follows the VICAIRE rule, « for each 5 mm. increase in the gradient, it is necessary to reduce the speed V_i by 5 km./h. ». M. VICAIRE considers that for gradients in excess of 30 mm. a speed of 25 km./h. could be retained without drawback, and that it is not necessary to exceed the above limits.

For the limitation of speeds on gradients, it is necessary to take into account the resistance of the track, the kind of locomotives and rolling stock and to take the lowest speed found in the two cases in question (see *Revue Générale des Chemins de Fer*, Jan., Feb. and March, 1880).

No. 2, right, shews these speeds in Graph No. III.

3. *On line No. 3* are the speeds given by the linear formula $V_i = 102 - 2i$, used, if our memory is correct, on the Swiss Railways.

Line No. 3 shews these speeds in Graph No. III.

4. *On line 4* are the speeds shewn by the formula :

$$Vi = 3\,625 - 300\sqrt{V}$$

(*) The first five lines of this table are the same as in Table No. II.

TABLE No. II.

Speeds V (lines 1-5) and radii ρ (lines 1'. 5') as functions of *i*.

Order.	Characteristic gradients *i*.											
	2	5	8	10	12	15	20	22	25	30	35	40
1.	...	100	80	72	67	60	50	45	40	32	27	20
2.	109	100	91	85	79	70	55	49	40	25	...	...
3.	98	92	86	80	78	72	62	58	52	42	32	22
4.	126	106	93	85	79	72	62	60	56	50	46	42
5.	110	103	95	90	85	78	65	61	54	40	35	30
Characteristic radii ρ :												
1'.	800	640	500	420	350	280	210	190	170	140	130	115
5'.	750	675	600	550	500	425	300	250	175	160	130	130

1. Curve No. 1 of the Austrian Railways.

2. Line No. 2 of Prof. VICAIRE $V = 115 - 3i$.

3. Line No. 3 of the Swiss Railways $V = 102 - 2i$.

4. Curve No. 4 $V \times i = 3\,625 - 300\sqrt{V}$,

for the round value of i $\sqrt{V} = \dfrac{-150 + \sqrt{22\,500 + 3\,625i}}{i}$

5. Lines No. 5 following the Author :

$V = 115 - 2.46i$ $\rho = 800 - 25i$ for main lines.

$V = 70 - i$ $\rho = 250 - 3i$ for secondary lines.

Speeds V and radii ρ correspond, vertically, to each gradient i.

or by the formula

$$\sqrt{V} = \frac{-150 + \sqrt{22\,500 + 3\,625i}}{i}.$$

We believe that this was proposed in the *Annales des Ingénieurs Civils de Gand* but the date and author are not known.

Curve No. 4 in Graph III shews these speeds.

5. *On line 5* are shewn the speeds calculated by the formulae :

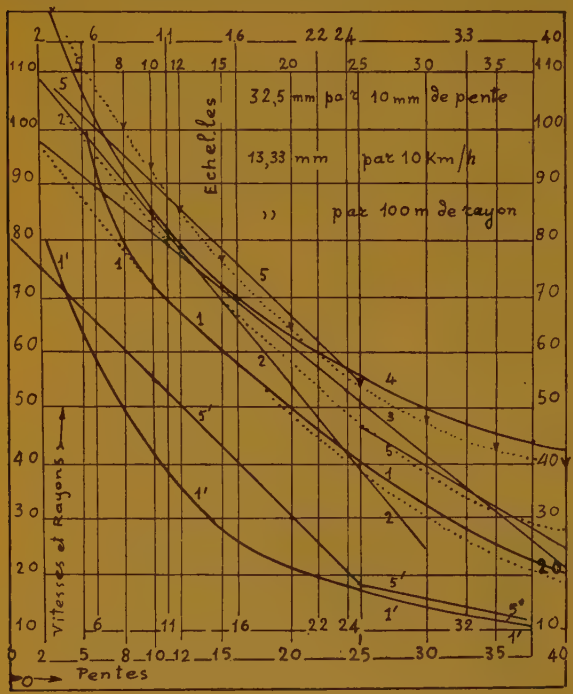
$V_i = 115 - 2.46i$ *for main lines*,

$V_i = 70 - i$ *for secondary lines*.

These speeds differ little from the re-

GRAPH No. III.

Equations V = $\varphi(i)$ and $\rho = \varphi_1(i)$ following the numerical tables II and III, lines 1, 2, 3, 4, 5, 1', 5', etc.



Explanation of French terms:

Vitesses et Rayons = Speeds and Radii.
Echelle 32.5 mm. par 10 mm. de pente = Scale 32.5 mm. per 10 mm. gradient.
Echelle 15.33 mm. par 10 km./h. = Scale 15.33 mm. per 10 km./h.
Echelle 15.33 mm. par 100 m. de rayon = Scale 15.33 mm. per 100 m. radii.

lative ones of the German Railways. They have been shewn on line 5 of Graph No. III.

6. *On line 6* is shewn the average of the speeds of each column for the first five lines.

These speeds differ little from those given by the formula $V = 0.03655i^2 - 3.693i + 117$.

7. *On line 7* are shewn the highest speeds indicated in the first five lines.

8. *On line 8* are shewn the lowest speeds indicated in the first five lines.

9. *On line 9* are shewn the highest speeds differing little from those of line 7 and calculated by the formula $V = 0.059i^2 - 4.69i + 134$ (*).

10. *On line 10* are shewn the lowest speeds, differing little from those of line 8 and calculated by the formula $V = 0.0307i^2 - 3.342i + 103$ (*).

11. *On line 11* are shewn the speeds calculated by the formula $V = 20.80\sqrt{29.80 - i}$ under No. 6 proposed by us in the Dec. 1933 *Bulletin of the International Railway Congress*. It will be noted that they differ little from Nos. 7' and 9'.

From all the formulae 1 to 12, as well as Table III, we see that on gradients i there are also *maximum high, low and average speeds*, according to the inclination; these speeds must be in accordance with the corresponding speeds calculated for the radii ρ.

12. *On line 12* are shewn the radii which correspond to the gradients i according to the formula $\rho = 1\,200 - 45i$ proposed in the above article to give the same speed as the gradient i.

13. *On line 13* are shewn the radii corresponding to the gradients i according to the formulae :

$\rho = 800 - 25i$ for main lines and i from 2 to 25 mm.

$\rho = 250 - 3i$ for secondary lines and i from 25 to 40 mm.

14. *On line 14* are shewn the radii corresponding to the formulae :

$\rho \geqslant 825 - 5i$ for $i = 0$ to 5 mm.

(*) See these curves in Graph No. IV.

TABLE No. III.

Speeds V (lines 1-11) and radii ρ (lines 12-14) as functions of *i*.

Order.	Characteristic gradients *i* :											
	2	5	8	10	12	15	20	22	25	30	35	40
1	...	100	80'	72'	67'	60'	50'	45'	40'	32	27'	20'
2	109	100	91	85	79	70	55	49	40'	25'	...	...
3	98'	92'	86	80	78	72	62	58	52	42	32	22
4	126v	106v	93	85	79	72	62	60	56v	50v	46v	42v
5	110	103	95v	90v	85v	78v	65v	61v	54	40	35	30
6''	110.5''	100.2''	89''	82.4''	77.6''	70.4''	58.8''	54.6''	48.4''	37.8''	35''	28.5''
7v	126v	106v	95v	90v	85v	78v	65v	61v	56v	50v	46v	42v
8'	98'	92'	80'	72'	67'	60'	50'	45'	40'	25'	27'	20'
9v	125v	112v	100v	93v	84v	77v	64v	60v	54v	47v	42v	41v
10'	97'	88'	79'	73'	68'	60'	49'	45'	39'	31'	23'	19'
11	110	104	97	93	88	80	65	58	46	...	...	...
12	1 100	975	840	750	660	525	300	210	75			
13	815	800	500	470	435	380	290	255	200	*radii.*		
14	750	675	600	550	500	425	300	250	175	160	145	130

$\rho \geqslant 100(13 - i)$ for $i = 5$ to 8 mm.

$\rho \geqslant 650 - 18i$ for $i = 8$ to 25 mm.

Formulae No. 13 correspond to the French system, No. 14 to the German system.

14. *a*) The French Est allows for its locomotives (see VANDERRYDT and MINSART, part 3, p. 237).

Speeds	50	45	40	35	30
for gradient	0—4	4—7	7—10	10—15	15—23

1. *For main lines :*

$\rho =$	1 200	1 100	1 000	900	800	700	600
$i =$	0	0	0	0	2	7	8

b) Correlation of the value ρ, *i*, V and h for the German system.

It is appropriate here to note the values and to shew their relation with the German system according to the « Hütte » manual (26th. edition, vol. 3, pp. 885 and 891). Comparison with the results given by the linear formulae which we propose for $V_i = \varphi(i)$ and $\rho = \varphi_1(i)$ shew clearly that we can use these formulae for the case *of the average maximum speeds*, from which differ little the *maximum speeds* of the German system ($V \approx 3.85\sqrt{\rho}$ instead of $V = 3.75\sqrt{\rho}$).

$V =$	120	120	120	115	110	100	95
$h =$	96	96	115	118	121	116	120
$\rho =$	500	400	300	250	225	200	180	150
$i =$	12	16	20	22	25	25	25	30
$V =$	85	75	65	60	55	50	45	40
$h =$	116	113	113	115	108	100	90	86

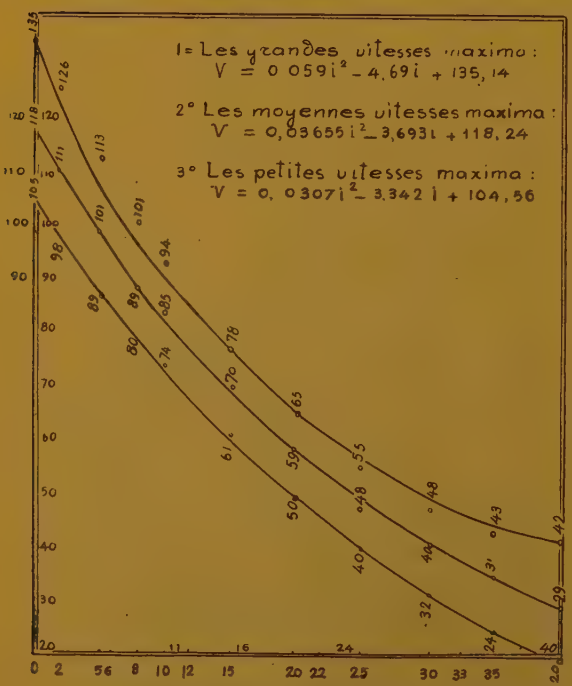
2. *For secondary lines :*

$\rho =$	200	180	150	120	100
$i =$	25	25	30	40	40
$V =$	50	45	40	30	25
$h =$	100	90	86	60	50

The superelevation h may be increased or decreased by 30 mm.; it is related according to the rules of the system for the smooth running of trains.

GRAPH No. IV.

High, medium and low maximum speeds as functions of i.


1° Les grandes vitesses maxima :
V = 0,0591 i² - 4,69 i + 135,14
2° Les moyennes vitesses maxima :
V = 0,03655 i² - 3,693 i + 118,24
3° Les petites vitesses maxima :
V = 0,0307 i² - 3,342 i + 104,56

Explanation of French terms :

Les grandes vitesses maxima = The high maximum speeds.
Les moyennes vitesses maxima = The medium maximum speeds.
Les petites vitesses maxima = The low maximum speeds.

c) Formulae $\rho = \varphi_1(i)$.

In order to have the same speed V with one or other of the two characteristics ρ and i of a speed section, it is necessary to have the equation $V_\rho = f(\rho) = V_i = \varphi(i)$, which sets up a relation $\rho = \varphi_1(i)$ between ρ and i.

The relationship of the sizes of ρ, i and V on the German Railways leads to the two equations :

$\rho = 800 - 25i$ for main lines and i from 2 to 25 mm.

$\rho = 250 - 3i$ for secondary lines and i from 25 to 40 mm.

Note. — The relationship above is not in accordance with alignment practice. We have in fact on the French systems :

$\rho \geqslant 825 - 5i$ for $i = 0$ to 5 mm.

$\rho \geqslant 100(13 - i)$ for $i = 5$ to 8 mm.

$\rho \geqslant 650 - 18i$ for $i = 8$ to 25 mm.

German practice requires smaller radii than those above for gradients between 2 and 6 mm. and larger radii for gradients of between 6 and 22 mm.

d) Rational use of the formulae.

On this matter we return to what has previously been said regarding the ap-

plication of the formula $V_\rho = f(\rho)$: formula $V_i = \varphi(i)$ is not applied for each value of i but only for the characteristic gradient i of the section; on less severe grades braking is less used; if the grades are more severe the speed is regulated according to the curves.

From the braking equation :

$$(w + \lambda r \pm i) \times S = (1.08 \text{ to } 1.10)\tfrac{1}{2}\frac{v^2}{g}$$

we see that the brake resistance $+ \lambda r$ is necessary if $w - i < 0$, i.e. if the resistance w (in kgr. per ton of the whole train) is less than the descent i (in mm./m.) :

We should thus have, as excessive slopes i_n, of a train composed of a 100 ton *Pacific* locomotive, 50 ton tender and a weight of 500 tons, comprising mainly vehicles of 40 and 56 tons (see Epinay, « Cours de Chemins de Fer de l'Ecole Nationale des Mines de Paris », Book IV, p. 146) running at different speeds, the following :

for V =	70	90	100	110	120	km.
$i_n >$	4.77	5.47	6	6.47	7.185	mm.

For a goods train composed of an engine and tender weighing about 70 tons in total and loaded wagons ($\approx$ 20 tons) to total 400 tons, running at a speed of 40 km./h., we should have a specific resistance of the full train of 4.27 kgr. or as excessive slopes, $i_n > 4.27$ mm. A speed of 20 km./h. would give $i_n > 2.91$ mm. It will be seen that the resistant work of braking $\lambda r \times S$ increases almost proportionately to the square of the speed v, and with the length of run s, with gradients more than 3 mm. for goods trains and 4 to 6 mm. for express and fast trains, that is to say, the full length of the track. It will therefore assist in reducing wear on tyres and rails, increase safety and comfort, *if speeds are not raised unduly,* particularly on long and steep gradients.

Chapter two.

D. 1. Influence of maximum speeds on curves and on rolling stock. — Table IV.

For the study of this question, we have included in Table No. IV the values of the accelerations a_e and a_i, and their relation $a_i : a_e$ for the case of a *speed section characterised by* $\rho = 500$ m. and $i = 12$ mm.

Calculations have been made for *high, medium and low maximum speeds,* and for various values of V_{min}, all by application of formulae 10, 12 and 13 and those giving the values of a_e and a_i.

2. Smooth running and wear of rails and tyres.

We have already taken :

$a_e \leqslant 0.40$ m./sec^2 *for smooth running.*

$0.40 < a_e \leqslant 0.60$ m./sec^2 *for permissible running.*

$a_e > 0.60$ m./sec^2 *for non-permissible running.*

The Italian Railways go to $a_e = 0.981$ m./sec^2, which gives rise to a disagreable blow towards the interior or exterior of the curve, according to whether there is an excess or deficiency of centrifugal force, in view of the constant inclination of the curve.

a) *Smooth running.*

From the values of a_e and a_i in Table No. IV, compared with the limits quoted above, we conclude :

1. *That high maximum speeds only give*

TABLE No. IV.

Reactions of the various maximum speeds.

Maximum speeds.		ρ	V	*h*	*d*	a_e	a_i	$a_i : a_e$
							m./sec²	
1° High maximum speeds.								
$V = 4.5\sqrt{\rho}$	$V_{min.}$ 30	500	101ᵛ	150	100	0.581	0.842	1.45
	40						0.734	1.26
	50						0.596	1.03
	60						0.425	0.73
	70						0.225	0.39
2° Medium maximum speeds.								
$V = 3.75\sqrt{\rho}$	$V_{min.}$ 30	500	84ᵛ	110	73.3	0.369	0.580	1.57
	40						0.472	1.28
	50						0.334	0.91
	60						0.163	0.44
3° Low maximum speeds.								
$V = 3\sqrt{\rho}$	$V_{min.}$ 30	500	67ᵛ	72	48	0.217	0.232	1.07
	40						0.224	1.03

smooth running, consequent upon $a_e = 0.581$ m./sec² and the values of a_i, for speeds $V_{min.} < 61.4$ km./h.

The superelevation $h = 150$ mm. allowed for speeds of 101 km./h. is that of the Italian system (see p. 265 of L. STABILINI's article mentioned in regard to formula No. 1).

2. *That medium maximum speeds give smooth running for speeds* $V_{min.}$ *above 45 km./h.;* local and stopping trains and goods trains are not smooth running in view of the excessive inclination in regard to their speed.

The superelevation h = 110 mm. which we have taken for these speeds is almost that of the German Railways, calculated by the formula $h = 8\frac{V^2}{\rho}$ (113 mm. instead of 110 mm.). On the French Midi $h = 100$ is allowed, PLM, 125 mm. and Est 134 mm.

3. *That low maximum speeds give very smooth running.*

The superelevation $h = 72$ mm. for this case has been calculated by the German formula. The French Midi also give the same superelevation $h = \frac{36}{\rho}$ for

V = 60 to 75 km./h.; the same system uses for medium maximum speeds $h = \frac{50}{\rho} = 100$ mm., and for high maximum speeds $h = \frac{66}{\rho} = 132$ mm. instead of $h = 150$ mm. as on the Italian system.

b) *Wear of rails and tyres.*

It is recognised that the wear increases greatly with the values of the accelerations a_e, a_i and consequently with the increase in V and the decrease in V_{min}. Trains of speeds higher than that used in the calculation of the incline overload the exterior rails and thrust them towards the outside of the curve, with wear on the top and lateral surfaces; whilst trains of speeds lower than that which corresponds to the incline allowed give rise to the same wear on the interior rails, and at the same time cause wear on the exterior rails by hunting. In the latter case, following the overloading of the interior rails, the tangential reaction of the inside driving wheels $P_i f$ is greater than that of the outside driving wheels $P_e f$ (since $P_i > P_e$). This gives rise to a pivoting movement about the front of the engine towards the outside and increases the lateral pressure on the outer rails by the fore outside tyres, with an increase in the angle.

In the two latter cases, wear of the wheel tyres follows closely the wear on the rails, and to the same degree.

3. Resistance to and forces opposing train running greatly increase with higher speeds. Influence of high speeds on safe running.

It is known that the formula for resistance to train running contains a factor proportional to the square of the speed of running, but it is often forgotten that the work absorbed by this resistance is proportional to the cube of the speed of running; moreover, it is very simple, since the resistant work is the product of the specific resistance $a + bv^2$ or $a + bv + cv^2$ and the corresponding speed v, that is :

$$T_w = (a + bv^2 \text{ or } a + bv + cv^2) \times v = av + bv^2 \text{ or } av + bv^2 + cv^2.$$

We may calculate separately the frontal resistance and the resistant work in *still air* to the running of a train.

We have already taken as the total specific resistance :

$$R_a = 0.006V^2 \times S \text{ kgr.}$$

the formula in which S is (in m^2) the largest transverse projection of the train (≈ 11 m^2 in the case of large modern locomotives). From this we get an absorbed work of $T'_a = 0.006V^2S \times \frac{V}{3.6} = 0.00166V^3S = 0.0183V^3$ in kgrm. for $S = 11$ m^2; expressed in steam H.P. of 75 kgrm. per second, this work will be given by the formula $T_a = V^3 \times 0.0002444$, still for $S = 11$ m^2 (*).

By the use of these formulae, Table No. V has been drawn up, the figures of which are reasonably explanatory as regards the increased resistance and also the resistant work in *still air* when the running speed increases. Special note has been taken of the resistances $R_a = 296$, 466 and 673 kgr. and the resistant work $T'_a = 74$, 145 and 252 H.P. in the cases of the *low, medium and high maximum speeds* V = 67, 84 and 101 km./h., noted in Table No. IV.

These values, as well as the differences

(*) This surface gives about 11.8 m^2 for the tender.

TABLE No. V.

Resistance and resistant work of the air.

V:	30	40	50	60	67	84	90	101	110	120	150
R_a kgr.	59	106	165	238	296	466	535	673	799	950	1 485
Differ.		47	59	73	58	170	69	138	126	151	535
T_a CV	7	16	31	53	74	145	178	252	325	422	825
Differ.		9	15	22	21	71	33	74	73	97	403
e_n min.	3.8	3.8	10.6	15.3	19	30	34.4	43.3	51.4	61.2	95.6
e_n max.	4.4	7.36	12.3	17.7	22	34.6	39.8	50.1	59.4	70.7	110.5
t_0''	5.7	8.3	9.7	14.3	16.7	23.2	25.8	30.5	35	40	54
s_0'' m.	29	40	88	129	172	308	369	497	617	795	1 355

Resistance of the air :

$$Ra = 0.006\,V^2 \times S \text{ kgr.}$$

Work absorbed :

$$Ta = 0.0000222...\,V^3 \times S \text{ CV.}$$
$$= 0.0002444...\,V^3 \text{ for } S = 11 \text{ m}^2.$$

Stopping time $t_0'' = 0.144\,V + 0.00158\,V^2$.

Stopping distance $s_0 = (0.02052 + 0.000273\,V)\,V^2$.

D between the successive values of R_a and T'_a clearly shew the great influence of the head resistance of *a still atmosphere*, without taking into account the intermediate frontal resistances, lateral resistances, the resistance arising from the pressure on the rear face of the train and finally the resistance of a wind more or less contrary to the direction of running.

It will be noted that the existence, as much as the cancellation of the kinetic energy of the trains $E_n = (1.08 \text{ to } 1.25) \times \frac{1}{2}mv^2$ are proportional to the speed of running.

The loss of time for starting and stopping trains also increases with the speed; thus for speeds of 67, 84 and 101 km./h. the French Est reckons as lost time, 1.87, 2.54 and 3.02 minutes for the whole of the two losses. These losses were calculated by the formula :

$$r' = 0.09V - 0.000333V^2 - 2.67$$

this being compiled from the factors relative to the Est system.

4. Safety of operation.

Train stops, scheduled or emergency, are always obtained by braking more or less heavily according to whether the speed of running is greater and the stopping distance reduced.

In applying the formulae

$$t_0'' = 0.144V + 0.00158V^2$$
$$s_0 = 90 - 4.51V + 0.085V^2 \text{ in m.}$$

or

$$s_0 = (0.02052 + 0.000273V)V^2 \text{ in m.}$$

we obtain the necessary time t_0'' to cancel V over the stopping distance s_0, as indicated in Table V (see work by M. Parodi and A. Tetrel « La Traction électrique et le chemin de fer », pp. 442-448).

The formulae in Table V shew that the times t_0'' and the distance s_0 increase in a large measure with V; this results in increased risk in the event of emergency stops.

The values t_0'' and s_0 are practically the same for the two hypotheses which could be made on the variation of the coefficient of friction of the brake blocks on the wheels : variation with the speed and time, or variation with the speed only : $f = f_0 : (1 + av)(1 + \beta t)$ or $f = f_0 : (1 + av)$.

5. Co-ordination of alignment for high, medium and low maximum speeds.

1. *High maximum speeds* demand very large radii; normally over 800 m., long straight alignments particularly between curves in reverse directions, descents of less than 5 mm./m., perfect transition curves both in longitudinal and plan profile, careful and constant maintenance and a solid track and bed. It is only with these conditions that the drawbacks of *high maximum speeds* can be overcome, it being taken for granted that the locomotives and rolling stock are also suitable : perfectly balanced engines with a leading bogie and normally three or possibly four driving axles, vehicles with bogies and as stable as possible in relation to their weight.

Railcars and *railcar sets* are better suited than trains with separate engines for *high maximum speeds*, consequent upon their suspension on driving bogies and their general structure (usually streamlined, etc.).

In the latter case, $V = 5\sqrt{\rho}$, or 11 % more than in the case of locomotives is proposed.

2. *Medium maximum speeds* presuppose the same exigencies for track and bed *as high maximum speeds*, with a tolerance in the case of gradients, which may fall to 8 mm./m. in exceptional cases, and curves which may at times be as sharp as 500 m.

Locomotives with leading bogies are not essential, at least up to 90 km./h. below this speed a guiding Bissel truck is an advantage : driving axles do not normally exceed four, but with increased stability can go up to five : bogie vehicles are not essential, but the wheelbase of those on two axles should not be below 4.5 to 5 m.

Driving bogies, with *single* railcars or coupled in *series*, are better than separate locomotives.

Between the *medium* and *high maximum speeds*, the speeds $V = 4\sqrt{\rho}$ of the English systems are indicated, provided that track of good standard, as used in England, is available.

3. *Low maximum speeds* are indicated for difficult alignments with radii up to 250 m. as the lower limit, occasionally 200 m., with gradients not exceeding 20 to 25 mm./m. In this case again, curves and transitions must be in a perfect state of maintenance. Especially it is necessary to have, between radii ρ and gradients i, a relation $i = \varphi(\rho)$ or $\rho = \varphi(i)$ so that the speed V, calculated by means of one or other of the characteristics, has the same value : without this relation, the smaller of the two speeds must be accepted without profiting from the greater.

6. Large and frequent variations of radius and gradient are a drawback from several points of view.

Railway alignment, just as much as roads, needs to be as economical as possible in earthwork and structures, from small aqueducts to large viaducts : the red longitudinal profile is constantly opposing the black, without even examining closely the transversal slope of the terrain. Great difficulties are also encountered in giving a free way to water at the required height for even rain water : it is often difficult to substitute an open aqueduct by an arched one, or to embed in concrete a metal culvert not sufficiently strong to support modern heavy or fast engines.

To achieve this economy the values of gradients and curve radii are changed at each step, in a more or less large degree, without regard to their length or their effect on locomotive power supplies from the point of view of speed restrictions on gradients, etc., thus there are large and small curves, steep and gradual gradients, curves and straights : in brief, a train must run in a constantly *varying* trajectory. The maximum gradient and the minimum curve y are often applied for the purpose of achieving insignificant economies.

This method of achieving economies is illogical : they are effected at the expense of efficiency of locomotives, speed of trains, smoothness of running, conservation of rails and rolling stock, and track life and maintenance. These facts have consequences inseparable from the remarks in this article concerning the fixing of speed $V = f(\rho)$ or $V_i = \varphi(i)$ on each *speed section*. Too often it is forgotten that the track is built as a corridor for rolling stock : *they are half the means of transport called a railway system* : they serve the rolling stock : 1) *as a solid support* having regard to the weight and speeds of the rolling stock; 2) *as compulsory trajectories or paths;* 3) *as corridors for the use, by means of the adhesion, of the work of the motors.*

Frequent and particularly large variations in curvature, gradient (position and length) act against the aims of 2 and 3 above and result in the consequences mentioned both as regards smooth running and wear on track and stock.

Locomotive efficiency, in particular, is lowered by frequent variations of gradient and curvature: maximum efficiency $E_a \times v_a$ is obtained if the motor works at a speed v'_a, which is near the optimum speed v_a, to which corresponds a tractive effort E_a at the tender drawbar hook.

The demands of motive units on rails, from all points of view (constitution, balance, operation, etc.) and also trailing vehicles, should be well understood by the Permanent Way Engineers : collaboration between Civil and Mechanical Engineers could produce noticeable improvement in future layouts in plan, longitudinal and transverse profile.

7. Necessity for greater opposition to the tendency for speed increases above 90-100 km./h. or 105 km./h. at most.

Locomotive and Permanent Way Departments must oppose the tendency towards higher and higher speeds. In effect, the Permanent Way Department is responsible for the maintenance in perfect condition of the track in relation to the static and dynamic loads to which it is subjected, as well as for the state of the plan and profile alignment. The speed factor is hostile to these two aims with the third power (see Tables IV and V) and increases the static pressures

perhaps beyond 50 % (see Parodi and Tetrel, pp. 281-284). The same Department will testify that with high speeds and especially with heavy trains, maintenance work becomes difficult, less lasting and more onerous.

Further, the Locomotive Department must provide Rolling Stock to run with safety and efficiency. To realise these two aims, the speed factor is, outside certain limits, variable between the motor and trailer vehicles, but more unfavourable as it mounts. The figures in Table V are revealing in this respect, although they deal only with head resistance and resistant work in *still air*.

In brief, the aims to be achieved by *the two Departments* are so essential that we should not hesitate to survey most strictly the application of speed regulations, and not concede increases in speed which in most cases are unjustified : and it *must not be forgotten that their close collaboration is absolutely necessary.*

Athens, 6th. August, 1947.

* * *

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[625 .113]

Transition curves,

by L. VUAGNAT,

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(*Bulletin Technique de la Suisse Romande, 10th. May, 1947.*)

At the beginning of the railway era, as soon as speeds began to rise, it became impossible to maintain curves in the same plane as the straight alignment.

The builders were compelled to give a cant to the track by raising the outer line of rails so as to balance the centrifugal force. To attain the difference in level a « superelevation incline » was inserted between the straight and the curve. Originally, this incline was placed entirely in the straight section, then half in the straight and half in the curve. Later the track was slewed slightly towards the centre of the curve at each end at the tangential point, over several lengths of rail, this being the first rough attempt at a transition curve. The success of this arrangement depended on the experience and « eye » of the foreman ganger.

Chavès and Nördling, engineers of the French Nord, in 1865 and 1867, told of the use of the cubic parabola

$$y = Bx^3$$

as a transition curve, shewing that the rectilinear superelevation incline required a transition curve the curvature of which increased proportionately to the length of the arc, and that the cubic parabola fulfilled this condition almost completely. Nördling shewed at the same time a method of picketing the curve which was both simple and practical, and which has continued to be used upon to recent years.

Consideration of superelevation inclines and the adaptation of the curvature of the track to the cant was therefore the beginning of the transition curve.

In the notes which follow, we shall review rapidly these considerations, giving accurate formulae, first taking the superelevation incline as a straight line, then as an analytical curve in order to generalise the question and to arrive at a satisfactory conclusion, from a dynamic as well as a geometric point of view, for the laying down of the track itself.

We may recall first of all, the formula for cant :

For a curve of fixed radius R and a speed V, the cant is given by the formula :

$$h = d_{max} = \frac{kV^2}{R} \qquad (1)$$

in which k is a coefficient which varies according to whether V is given in m./sec. or in km./h. It also varies, if we wish to obtain the theoretical cant, that is the complete balancing of the centrifugal force, as opposed to practice, in which part of the centrifugal force is not compensated. (Not to obscure this note, we shall not expound this formula.)

For a curve of variable radius ρ, we have

$$d = \frac{kV^2}{\rho} \qquad (2)$$

For simplicity, it may be taken that the cant is obtained by raising the outer line of rail and not as prescribed by the

Swiss Federal Railways, half by raising the outer rails and half by lowering the inner rails.

1. Transition curve with rectilinear superelevation incline.

We have :

$$\frac{1}{m} = \frac{h}{L} \qquad (3)$$

$$d = \frac{1}{m}\, s = \frac{kV^2}{\rho} = \frac{h}{L}\, s \qquad (4)$$

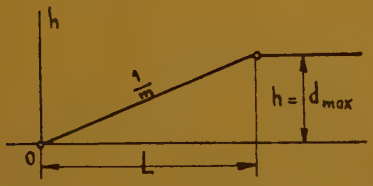
$$h = \frac{kV^2}{R} \qquad (5)$$

from which we take :

$$\frac{1}{\rho} = \frac{1}{RL}\, s = cs \qquad (6)$$

if we let

$$c = \frac{1}{RL} \qquad (7)$$



If the angle $d\Theta$ contained between two radii of curves infinitely adjacent is :

$$\rho d\Theta = ds = \frac{d\Theta}{cs} \qquad (8)$$

$$d\Theta = csds \qquad (9)$$

By integration it becomes :

$$\Theta = \frac{c}{2}s^2 \qquad (10)$$

However, we have, as rectangular co-ordinates :

$$dx = \cos\Theta\, ds \quad (11) \qquad dx = \cos\frac{c}{2}s^2 ds \quad (13)$$

from which

$$dy = \sin\Theta\, ds \quad (12) \qquad dy = \sin\frac{c}{2}s^2 ds \quad (14)$$

and by integration

$$x = \int_0^s \cos\frac{c}{2}s^2 ds \qquad (15)$$

$$y = \int_0^s \sin\frac{c}{2}s^2 ds \qquad (16)$$

To calculate these integrals, we may develop the sine and cos in series. This can be done because, for the values of s which we use, the series are rapidly convergent.

$$\sin\frac{c}{2}s^2 = \frac{c}{2}s^2 - \left|\frac{c}{2}\right|^3 \frac{s^6}{3!} + \left|\frac{c}{2}\right|^5 \frac{s^{10}}{5!} \cdots \qquad (17)$$

$$\cos\frac{c}{2}s^2 = 1 - \left|\frac{c}{2}\right|^2 \frac{s^4}{2!} + \left|\frac{c}{2}\right|^4 \frac{s^8}{4!} \cdots \qquad (18)$$

from which, by integration of the two series :

$$x = \left[s - \left|\frac{c}{2}\right|^2 \frac{s^5}{10} + \left|\frac{c}{2}\right|^4 \frac{s^9}{216} \cdots\right]_0^s \qquad (19)$$

$$y = \left[\frac{c}{2}\frac{s^3}{3} - \left|\frac{c}{2}\right|^3 \frac{s^7}{42} + \left|\frac{c}{2}\right|^5 \frac{s^{11}}{1320} \cdots\right]_0^s \qquad (20)$$

and giving to c the values taken from (7)

$$x = \left[s - \frac{s^5}{40R^2L^2} + \frac{s^9}{3456\,R^4L^4} \cdots\right]_0^s \qquad (21)$$

$$y = \left[\frac{s^3}{6\,RL} - \frac{s^7}{336\,R^3L^3} + \frac{s^{11}}{42240\,R^5L^5} \cdots\right]_0^s \qquad (22)$$

We immediately see that if we take the first term of each series, we return to the equation

$$y = Bx^3 \text{ where } B = \frac{1}{6RL}.$$

Given that s varies between 0 and L

$$s = nL$$

whence n varies between 0 and 1.

This becomes :

$$x = nL - n^5\frac{L^3}{40R^2} + \frac{n^9L^5}{3456R^4} \quad (23)$$

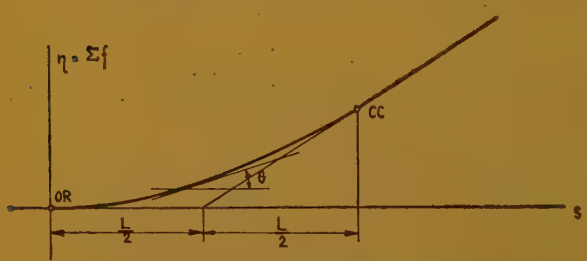
$$y = \frac{n^3L^2}{6R} - \frac{n^7L^4}{336R^3} + \frac{n^{11}L^6}{42240R^5} \quad (24)$$

and taking the first two terms

$$x = nL\left[1 - \frac{n^4L^2}{40R^2}\right] \quad (25)$$

$$y = \frac{n^3L^2}{6R}\left[1 - \frac{n^4L^2}{56R^2}\right] \quad (26)$$

In considering transition curves by the angle diagram method (1) we have the following construction :



from which $\text{tg}\Theta = \frac{1}{\rho} = ks \quad k = \frac{1}{RL}$ (27)

$$\frac{d\eta}{ds} = ks \quad d\eta = ksds \quad (28)$$

By integration, we have :

$$\eta = \Sigma f = \frac{k}{2}s^2 \quad (29)$$

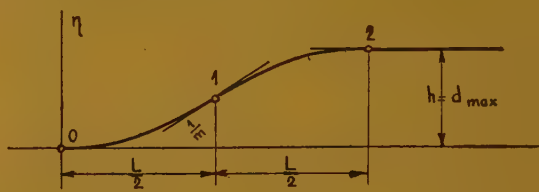
$$\eta = \frac{L}{2R}n^2 \quad (30) \quad \text{with } n = \frac{s}{L}.$$

In practice, the unit of measurement being Δl, the second member of equation No. 30 will have to be multiplied by Δl.

2. Transition curve with varying superelevation slope.

Instead of a constantly increasing cant, we may provide a variable increase.

Subject the superelevation incline, which will be the curve representing the cant, to the following conditions :



to be tangential to the point 0 at the axis of the ξ and to the point 2 to a // at the axis of the ξ. The points 0, 1 and 2 will be the points of inflexion.

We reach a function $\eta = \varphi(\xi)$.

This function has three roots :

$$\xi, \left(\xi - \frac{L}{2}\right), (\xi - L)$$

in addition $\varphi'(\xi) = 0$ for $\xi = 0$ and $\xi = L$.

We therefore get :

$$\frac{d^2\eta}{d\xi^2} = A.\ \xi.\left(\xi - \frac{L}{2}\right).(\xi - L)$$

$$= A\left(\xi^3 - \frac{3L}{2}\xi^2 + \frac{L^2\zeta}{2}\right) \quad (31)$$

$$\frac{d\eta}{d\xi} = A.\left(\frac{\xi^4}{4} - \frac{L\xi^3}{2} + \frac{L^2\xi^2}{4}\right) \quad (32)$$

$$\eta = A.\left(\frac{\xi^5}{4.5} - \frac{L\xi^4}{8} + \frac{L^2\xi^3}{12}\right) \quad (33)$$

(1) *Bulletin technique de la Suisse romande*, Nos. of the 17th and 31st August 1946. E. DÉCOMBAZ, Engineer : « Exposé succinct de la méthode de rectification des courbes de chemins de fer d'après la mesure des flèches ».

Let us determine the coefficient A.

For $\xi = L$ we have $\eta = h$.

$$h = A\left(\frac{L^5}{20} - \frac{L^5}{8} + \frac{L^5}{12}\right) = A\,\frac{L^5}{120} \qquad (34)$$

$$A = \frac{120\,h}{L^5} \quad \frac{1}{m} = \operatorname{tg}\alpha_1 = \frac{15\,h}{8\,L} \qquad (35)$$

$$\eta = \frac{30\,h}{L^5}\left(\frac{\xi^5}{5} - \frac{L\xi^4}{2} + \frac{L^2\xi^3}{3}\right) \qquad (36)$$

If we make $\xi = s$ and $\eta = d$, we get, by replacing h and d by their values taken from (1) and (2) :

$$kV^2\frac{1}{\rho} = kV^2\,\frac{30}{RL^5}\left(\frac{S^5}{5} - \frac{LS^4}{2} + \frac{L^2S^3}{3}\right) \qquad (37)$$

On the other hand, as we have seen $\rho d\Theta = ds$

$$\Theta = \int_o^s \frac{1}{\rho}\,ds = \frac{30}{RL^5}\int_o^s \left(\frac{s^5}{5} - \frac{Ls^4}{2} + \frac{L^2s^3}{3}\right) ds$$

$$\Theta = \frac{1}{2\,RL^5}\left(2s^6 - 6Ls^5 + 5\,L^2s^4\right)_o^s \qquad (38)$$

and as in the preceding case

$$dx = \cos\Theta ds \quad \cos\Theta = 1 - \frac{\Theta^2}{2!} + \frac{\Theta^4}{4!} \cdots$$

$$dy = \sin\Theta ds \quad \sin\Theta = \Theta - \frac{\Theta^3}{3!} + \frac{\Theta^5}{5!} \cdots$$

These series being very rapidly convergent for the values of Θ used, we will take only the first two terms of each series. By integration, we get :

$$x = \int_o^s ds - \int_o^s \frac{1}{(2\,RL^5)^2}\,\frac{(2s^6 - 6\,Ls^5 + 5\,L^2s^4)^2}{2!}\,ds \qquad (39)$$

$$y = \int_o^s \frac{1}{2\,RL^5}(2\,s^6 - 6\,Ls^5 + 5\,L^2s^4)\,ds - \int_o^s \frac{1}{2\,RL^5}\,\frac{(2\,s^6 - 6\,Ls^5 + 5\,L^2s^4)^3}{3!}\,ds \qquad (40)$$

$$x = \left(s - \frac{1}{2!}\,\frac{1}{4\,R^2L^{10}}\right)\left[\frac{4}{13}\,s^{13} - 2\,Ls^{12} + \frac{56}{11}L^2s^{11} - 6\,L^3s^{10} + \frac{25}{9}L^4s^9\right]_o^s \qquad (41)$$

$$y = \frac{1}{2\,RL^5}\left(\frac{2}{7}s^7 - Ls^6 + L^2s^5\right) - \frac{1}{3!\;8\,R^3L^{15}}\left[\frac{8}{19}\,s^{19} - \frac{72}{18}\,Ls^{18} + \frac{276}{17}L^2s^{17} - \frac{576}{16}L^3s^{16} + \frac{690}{15}L^4s^{15} - \frac{450}{14}L^5s^{14} + \frac{125}{13}L^6s^{13}\right]_o^s \qquad (42)$$

For $s = L$ we get

$$x = L - \frac{L^3}{8\,R^2}\left[\frac{4}{13} - 2 + \frac{56}{11} - 6 + \frac{25}{9}\right] = L\left[1 - 0{,}022\left(\frac{L}{R}\right)^2\right] \qquad (43)$$

$$y = \frac{L^2}{2R}\,\frac{2}{7} - \frac{L^4}{48R^3}\left[\frac{8}{19} - \frac{72}{18} + \frac{276}{17} - \frac{576}{16} + \frac{690}{15} - \frac{450}{14} + \frac{125}{13}\right] = \frac{L^2}{7R}\left[1 - 0.0268\left(\frac{L}{R}\right)^2\right] \qquad (44)$$

As in the preceding case, we make $s = nL$, where $0 > n > 1$

$$x = L\left\{n - \left(\frac{L}{R}\right)^2\left[\frac{n^9}{8}\left(\frac{4n^4}{13} - 2n^3 + \frac{56}{71}\,n^2 - 6n + \frac{25}{9}\right)\right]\right\} \qquad (45)$$

n	0	1	2	3	4	5	6	7	8	9	Δ	n
,080	,000185	,000195	,000204	,000214	,000225	,000235	,000246	,000257	,000269	,000281	..11	,080
,090	,000293	,000305	,000320	,000334	,000348	,000362	,000377	,000393	,000409	,000425	..14	,090
,100	,000442	,000459	,000476	,000494	,000513	,000533	,000553	,000574	,000595	,000616	..19	,100
,110	,000637	,000660	,000684	,000709	,000734	,000759	,000784	,000809	,000835	,000862	..27	,110
,120	,000892	,000922	,000952	,000983	,001013	,001045	,001078	,001111	,001145	,001180	..33	,120
,130	,001215	,001251	,001288	,001325	,001363	,001403	,001444	,001485	,001526	,001568	..40	,130
,140	,001612	,001657	,001702	,001748	,001796	,001844	,001892	,001943	,001995	,002046	..48	,140
,150	,002097	,002152	,002207	,002263	,002319	,002376	,002935	,002495	,002556	,002618	..57	,150
,160	,002680	,002745	,002811	,002877	,002943	,003012	,003081	,003154	,003227	,003299	..70	,160
,170	,003372	,003448	,003524	,003604	,003684	,003764	,003846	,003929	,004011	,004098	..80	,170
,180	,004184	,004270	,004360	,004452	,004542	,004635	,004732	,004828	,004924	,005024	..95	,180
,190	,005124	,005224	,005326	,005432	,005538	,005648	,005758	,005868	,005979	,006092	.108	,190
,200	,006207	,006324	,006443	,006562	,006684	,006807	,006932	,007057	,007185	,007314	.125	,200
,210	,007446	,007579	,007712	,007847	,007985	,008126	,008267	,008408	,008551	,008698	.140	,210
,220	,008845	,008995	,009149	,009303	,009458	,009614	,009774	,009935	,010095	,010260	.160	,220
,230	,010425	,010593	,010764	,010935	,011109	,011285	,011464	,011643	,011826	,012009	.177	,230
,240	,012194	,012379	,012571	,012763	,012997	,013151	,013348	,013548	,013749	,013954	.195	,240
,250	,014159	,014368	,014570	,014790	,015004	,015221	,015441	,015660	,015885	,016111	.220	,250
,260	,016339	,016566	,016759	,017033	,017272	,017511	,017749	,017993	,018240	,018487	.240	,260
,270	,018736	,018990	,019245	,019504	,019762	,020024	,020289	,020554	,020825	,021096	.263	,270
,280	,021367	,021646	,021926	,022206	,022490	,022779	,023068	,023359	,023652	,023949	.290	,280
,290	,024248	,024549	,024853	,025157	,025468	,025779	,026096	,026412	,026730	,027053	.315	,290
,300	,027378	,027706	,028035	,028369	,028705	,029043	,029384	,029727	,030072	,030422	.340	,300
,310	,030773	,031128	,031486	,031846	,032208	,032576	,032941	,033313	,033687	,034063	.366	,310
,320	,034444	,034825	,035210	,035600	,035990	,036386	,036783	,037182	,037583	,037989	.395	,320
,330	,038397	,038808	,039223	,039640	,040061	,040483	,040912	,041340	,041773	,042207	.430	,330
,340	,043086	,043086	,043531	,043977	,044427	,044882	,045338	,045797	,046260	,046724	.456	,340
,350	,047189	,047665	,048141	,048618	,049101	,049584	,050073	,050563	,051057	,051553	.484	,350
,360	,052054	,052559	,053066	,053576	,054087	,054604	,055123	,055646	,056173	,056701	.520	,360
,370	,057234	,057769	,058307	,058850	,059394	,059943	,060494	,061051	,061611	,062173	.550	,370
,380	,062736	,063307	,063879	,064455	,065031	,065615	,066200	,066789	,067379	,067976	.585	,380
,390	,068574	,069179	,069784	,070391	,071004	,071621	,072338	,072864	,073490	,074119	.617	,390
,400	,074752	,075389	,076028	,076668	,077318	,077969	,078624	,079288	,079941	,080607	.655	,400
,410	,081827	,081945	,082627	,083300	,083982	,084666	,085355	,086048	,086744	,087445	.685	,410
,420	,088148	,088854	,089566	,090281	,090996	,091717	,092444	,093172	,093902	,094638	.724	,420
,430	,095378	,096120	,096868	,097676	,098370	,099127	,099887	,100652	,101420	,102194	.765	,430
,440	,102968	,103747	,104529	,105316	,106107	,106899	,107697	,108498	,109303	,110110	.795	,440
,450	,110921	,111738	,112556	,113380	,114207	,115036	,115870	,116709	,117549	,118394	.835	,450
,460	,119245	,120096	,120951	,121812	,122675	,123542	,124415	,125289	,126167	,127049	.873	,460
,470	,127935	,128829	,129719	,130619	,131520	,132423	,133320	,134241	,135157	,136076	.906	,470
,480	,136999	,137927	,138856	,139792	,140728	,141672	,142619	143566	,144521	,145477	.945	,480

| | | |

$$y = \frac{L^2}{R}\left\{\frac{n^5}{2}\left(\frac{2n^2}{7} - n + 1\right) - \left(\frac{L}{R}\right)^2\left[\frac{n^{13}}{48}\frac{8}{9}n^6 - \frac{72}{18}n^5 + \frac{276}{11}n^4 - \frac{576}{16}n^3 + \frac{690}{15}n^2 - \frac{450}{14}n + \frac{125}{13}\right]\gamma_n\right\} \quad (46)$$

$$x = L\left[n - \left(\frac{L}{R}\right)^2 \alpha_n\right] \quad (47)$$

$$y = \frac{L^2}{R}\left[\beta_n - \left(\frac{L}{R}\right)^2 \gamma_n\right] \quad (48)$$

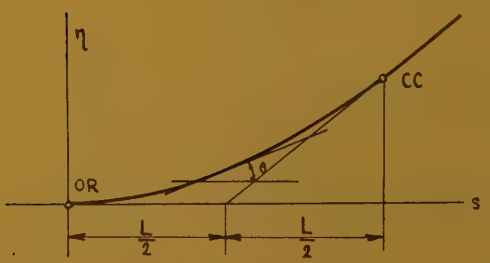
From formula (36) we take the value of the cant, making $\xi = nL$.

We get :

$$d = \frac{C_1}{[5n^4(1{,}2n^2 - 3n + 2)]}\,h \quad (49)$$

If, as in the preceding case, we consider this new curve by the aid of the angle diagram method, we get the following construction :

$\text{tg}\Theta = \frac{1}{\rho} = \frac{d\eta}{ds}$ taking $\frac{1}{\rho}$ from (37)



From this :

$$\frac{1}{\rho} = \frac{d\eta}{ds} = \frac{30}{RL^5}\left(\frac{s^5}{5} - \frac{Ls^4}{2} + \frac{L^2s^3}{3}\right) \quad (50)$$

$$\int d\eta = \frac{30}{RL^5}\int\left(\frac{s^5}{5} - \frac{Ls^4}{2} + \frac{L^2s^3}{3}\right)ds \quad (51)$$

$$\eta = \frac{30}{RL^5}\left(\frac{s^6}{30} - \frac{Ls^5}{10} + \frac{L^2s^4}{12}\right) \quad (52)$$

TABLE 1.

n	1	0,9	0,8	0,7	0,6	0,5	0,4	0,3	0,2	0,1
α_1	0,0220	0,011900	0,00567	0,00234	0,00078	0,00020	0,0002	—	—	—
β_1	0,142857	0,097853	0,062727	0,036975	0,019551	0,005351	0,003887	0,000832	0,000130	0,000005
γ_1	0,002685	0,001191	0,000411	0,000154	0,000025	0,000004	—	—	—	—

TABLE 2.

n	1,00	0,95	0,90	0,85	0,80	0,75	0,70	0,65	0,60	0,55	0,50	0,45	0,40	0,35	0,30	0,25	0,20	0,15	0,10	0,05
C_1	1	0,999	0,991	0,972	0,942	0,896	0,837	0,765	0,683	0,593	0,500	0,407	0,317	0,235	0,163	0,104	0,058	0,027	0,009	0,001

If we make $S = nL$

$$\eta = \frac{L}{2R} \cdot n^4 (2n^2 - \overset{C_2}{6n} + 5) \qquad (53)\ (1)$$

In practice, the unit of measurement being Δl, the second member of equation (53) will have to be multiplied by Δl.

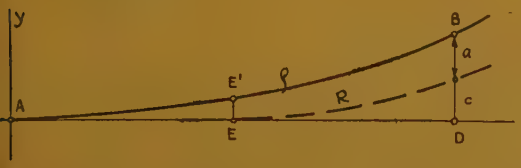
* * *

A table is given which allows the calculation of the coefficients C_2 (see pp. 556-557).

n being given to 3 decimal places instead of the required 5, the last 2 can easily be obtained by interpolation, with a sufficient degree of accuracy, by means of the differences shewn in the last column. This operation may be done accurately by calculation (2).

Displacement to the tangent circle.

Taking the first term of (43) and (44) we have :



$$AB = L \approx AD \qquad BD = \frac{L^2}{7R}$$

$$ED \approx \frac{L}{2} \qquad CD = \frac{L^2}{8R}$$

$$a = BD - CD = \frac{L^2}{R}\left(\frac{1}{7} - \frac{1}{8}\right) = \frac{L^2}{56R}$$

Transition curves with graduated superelevation inclines offer several advantages both for operation and for track laying.

From the operating point of view, it avoids oscillation of vehicles, both when passing from the straight to the transition and from the transition to the full curve. (See G. MARIÉ, « *Traité de la stabilité du matériel de chemin de fer* », p. 207.)

It is in order to avoid these oscillations that attempts are made to lengthen the present transition curves. Up to the present the Swiss Federal Railways have used the formula $L = 0.7V$ where V is in km./h. and L in m. but the possibility of making $L = V$ is being considered.

This is where the graduated superelevation incline achieves its advantages.

In effect, the displacement of the tangent circle is, for the present transition curve, from $a = \frac{L^2}{24R}$, whilst for the second curve, we have

$$a = \frac{L^2}{56R}$$

Let us see what these two formulae give in an actual case :

Take

$V = 125$ km./h.

$R = 830$ m. (Operating rules, p. 11) ;

$h = 106$ mm. (General regulation No. 230, p. 37).

1. *Transition curve with rectilinear superelevation incline :*

$$L = V = 125 \text{ m.} \qquad a = \frac{125^2}{24.830} = 0.785 \text{ m.}$$

(1) In the case of a transition curve between two curves of radii R_1 and R_2 we get :

$$\eta = L\left[\frac{1}{2}\left(\frac{1}{R_2} - \frac{1}{R_1}\right)[n^4(2n^2 - 6n + 5)] + \frac{n}{R_1}\right] \qquad (54)$$

(2) If, instead of the angle diagram, we use the deflection diagram, it would suffice to use Table 2, which gives the coefficient C_1, and we should have :

$$f = f_{max.}\, C_1$$

2. *Transition curve with graduated superelevation incline :*

$$L = \frac{0.24\,Vh}{32} = \frac{0.24 \cdot 125 \cdot 106}{32} = 100 \text{ m.}$$

If we make $L = 110$ m. which is still more favourable, we shall have :

$$a = \frac{110^2}{56.830} = 0.260 \text{ m.}$$

Transition curves on the writer's railway have at present a maximum length of 70 m.

In this case we have :

$$a = \frac{70^2}{24.830} = 0.243 \text{ m.}$$

We see that in the first case, we can effect a movement of

$$0.785 - 0.243 = 0.542 \text{ m.,}$$

which is not possible with present installations, whilst in the second case we have :

$$0.260 - 0.243 = 0.017 \text{ m.}$$

[621 .431 .72 & 625 .251]

Engine exhaust brake for railcars,

by J.L. KOFFMAN,

Dipl.-Ing. M.S.A.E., A.M.I. Loco. E.

The rapid technical development of diesel engined railcars and locomotives witnessed in the period between the two wars has now given place to a wider utilisation of such vehicles of what may be described as well established orthodox designs. In common with so many other present-day developments, it required some fifteen to twenty years to move the designs out of the region of high prices and limited utility into the region of reasonably low prices and very wide utility, but in accordance with the law of diminishing returns more development work will be required in the future for relatively small gains.

The development so far may be represented by the curve Fig. 1, the present position being indicated by the point 1. From this the development can move towards point 2 or 3, since a move in either direction would be progressive, but it should not move into the area bounded by the dotted line. If we consider for a moment the direction of future development it would appear that whilst the engine, and to a somewhat lesser degree the transmission has reached a high state of development much remains to be done so far as the development of light-weight railcar bodies and bogies is concerned. But apart from this there seems to be a further possibility towards improvement, one that on the first glance seems to be a rather obvious one, particularly since it was with us all the time and yet was not considered seriously at all — or at least so it appears from the scrutiny of the literature. That possibility is indicated by the question : « Why not use the

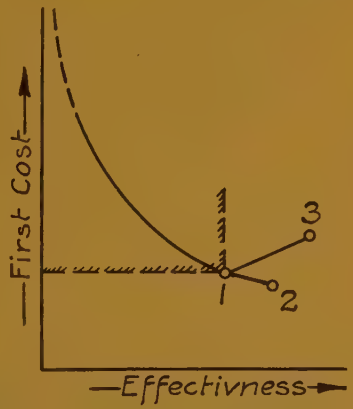


Fig. 1. — Technical progress.

engine for braking the vehicle since if driven from the axles it would act as a compressor in a somewhat similar way as tramcar motors can be made to act as dynamos and thus made to bring the car almost to a standstill ? »

This article is an attempt to answer this question, an attempt to show the possibility of moving the curve of progress from point 1 to point 2.

At first, we must consider how much

brake power can be made available by four-stroke engines without a major modification, since it is presumed that only a minor modification can be tolerated to make this proposition an attractive one. To do this we must know the losses encountered in diesel engines. Whilst the relevant information is of a somewhat scanty quantitative nature it is nevertheless sufficiently complete to enable us to form a clear qualitative picture.

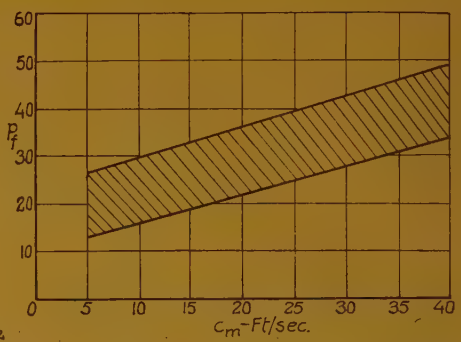


Fig. 2. — Range of mean friction pressures for diesel and petrol engines.

Whilst it is possible to express the friction, etc., losses of an engine in H.P. this would also mean a consideration of engine speed and swept volume, thus making a direct comparison between different engines somewhat involved. To avoid this it will be preferable to express the losses as mean friction pressure (M.F.P.) p_f expressed in lb. per sq. in. similarly to the mean effective pressure p_e so that the mechanical efficiency of the engine will be expressed as :

$$\eta_m = \frac{p_e}{p_e + p_f} \quad . \quad . \quad . \quad . \quad (1)$$

The values of p_f ascertained for a considerable number of engines having p_e values of up to 130 to 155 lbs. per sq. inch are shown, in Fig. 2., diesel engines tending toward the higher, and petrol engines towards the lower limits. These losses are caused by a number of contributing factors the magnitude of which will be indicated in the following on the basis of comprehensive tests carried out by ULLMAN (1) (*) and KRESS (2). The first series of tests were carried out on a single-cylinder engine having a bore and stroke of 80 and 129.7 mm. respectively and a crank/connecting rod ratio of ¼. The inlet valve was 40 mm. in diameter, the lift being 8.15 mm. It opened at top centre and closed 45° past bottom centre, whilst the exhaust valve, which had a diameter of 36 mm. and a lift of 8.25 mm. opened 39° before bottom centre and closed 5° after top centre. The depen-

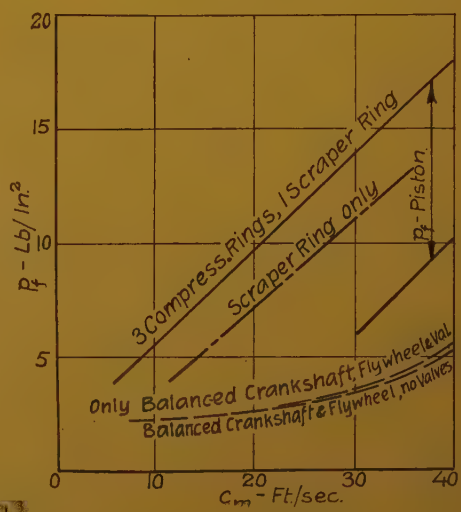


Fig. 3. — Mean friction pressure as a function of piston velocity.

(*) Numbers between parentheses refer to the Bibliography at the end of the article.

dence of p_f upon c_m due to a number of contributing elements is shown in Fig. 3 from which it will be noted that the losses due to the valve operating gear are negligible. The crankshaft can be assumed to run fully floating in oil. For this condition the friction losses are given by :

$$N_f = \frac{P \cdot n \cdot d \cdot \pi \cdot \mu}{550 \cdot 60} \text{ [H.P.]} \quad . \quad . \quad (2)$$

where :

P = load (lb.).

n = engine R.P.M.

d = journal diameter (ft.).

$\mu = 3.8\sqrt{\frac{z \cdot \omega}{p_m}}$ [according to FALZ (3)].

$\omega = 0.105n$ = rotational velocity (l./sec.).

z = mean absolute viscosity (lb. sec²/ft²).

$p_m = \frac{P}{d \cdot l} = \frac{P}{d^2(l : d)}$ = mean pressure (lb./ft²).

l = journal length (ft.).

Consequently :

$$N_f = \frac{d^2}{8\,500}\sqrt{P \cdot n^2 \cdot z(l : d)} \text{ [H.P.]} \quad (2a)$$

or

$$N_f \approx d^2\sqrt{Pn^2z} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2b)$$

For geometrically similar engines having identical ratios of H/D, d/D, etc., the thermal loads can be assumed to be proportional to the mean piston velocity c_m. Since the weight of the rotating and reciprocating masses is approximately proportional to $D^{2.6}$ we have

$$P \approx c_m^2 D^{1.6} \qquad N_f \approx c_m^{1.5} \cdot D^{2.3} \cdot n \cdot \sqrt{z}$$

and

$$P_f \approx \frac{c_m^{1.5}}{D^{0.7}} \cdot \sqrt{z} \quad . \quad . \quad . \quad . \quad (3)$$

i.e., for the same values of c_m an increase of D results in reduced p_f values.

Generally for big end bearings $d = 0.65D$ and $l = 0.45D$ for high speed in line engines with D<140 mm., whilst for V engines $d = 0.7D$ and $l = 0.48D$. The values for crankshaft bearing are $d = 0.76D$ to $0.69D$ and $l = 0.46D$ to $0.42D$ for in line engines and $d = 0.75$ and $l = 0.41D$ for V type engines.

Fig. 4

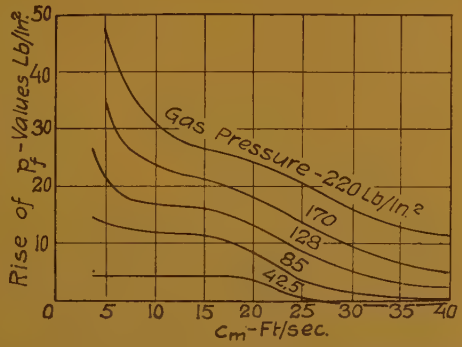


Fig. 4. — Effect of gas pressure upon increase of M.F.P.

The losses due to piston as well as due to piston ring friction are also indicated in Fig. 3 for zero pressure in the cylinder. The influence of the gas pressure upon the friction losses due to bearings and piston rings is shown in Fig. 4, from which it will be noted that the value of the increases of losses due to gas pressure is reduced as the piston velocity in-

creases. This is due to the predominant influence of the mass forces at the higher speeds, the influence of the gas pressure being mainly of importance at low speeds.

The friction losses which are encountered at constant gas pressures, maintained during the compression and expansion stroke only, are shown by the three straight lines of Fig. 5. Here again it will be noted that the influence of the pressure values becomes less and less pronounced as the piston velocity increases, since at higher velocities the influence of the mass forces predominates, as may well be proved analytically. The line obtained with a gas pressure of 140 lb. per sq. inch would approximately correspond to the line of friction losses encountered when driving an average modern high-speed diesel engine as a compressor whilst the corresponding losses encountered by the engine developing full power output would be similar to those obtained at a gas pressure of 225 lbs. per sq. inch.

Both the pumping and friction losses are also shown in Fig. 5 for the engine with a compression ratio of 1 to 16 when motored with the fuel supply cut off. The indicator diagram obtained with the motored engine is shown in Fig. 6. The difference of pressures encountered during the compression and expansion stroke is due to the cooling of the compressed air during the expansion stroke. The heat transfer coefficient α

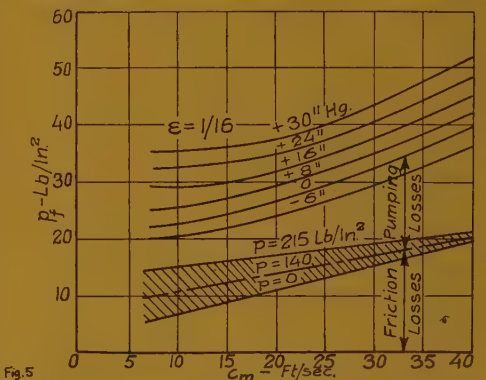


Fig. 5. — Effect of gas pressure on M.F.P. of engine.

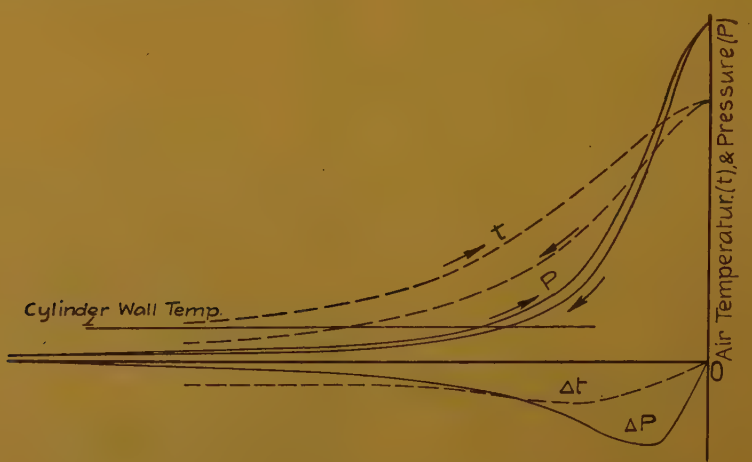


Fig. 6. — Work done during compression and expansion stroke.

for low speed (50 to 200 r.p.m.). Diesel engines can be estimated according to NUSSELT (4) as :

$$\alpha = 0.99 \sqrt[3]{p_m^2 \cdot T_g}\,(1 + 1.24 c_m) \left[\frac{kcal}{m^2 \cdot h \cdot °C.}\right] \quad . \quad . \quad . \quad (4)$$

where :

p_m = mean gas pressure [*at*].
T_g = mean gas temperature [°K].
c_m = mean piston velocity [m./sec.].

whilst according to EICHELBERG (5) :

$$\alpha = 2.1 \cdot \sqrt[3]{c_m} \cdot \sqrt{p_m T_g} \left[\frac{kcal}{m^2 \cdot h \cdot °C.}\right] \quad . \quad . \quad . \quad (5)$$

The latter equation has been well substantiated for c_m values of up to about 27 ft. per sec. by IBRAHIM (6), whilst NUSSELT's equation gives too low values for $c_m < 15$ ft. per sec. and too high ones for $c_m > 15$ ft./sec.

The amount of heat transferred during the time interval 1-2 is given by :

$$Q_{1-2} = \alpha \cdot 0.0001 A_{1-2} \cdot \Delta\sigma\Delta t_{1-2} \; [kcal] \; (6)$$

where :

A = mean cylinder area at time interval 1-2 [m.].

$\Delta\delta$ = time interval 1-2 = $\frac{\Delta\varphi}{6n}$ [hr.].

$\Delta\varphi$ = crankshaft displacement [°].

Δt = temp. difference (Tg — $Twall$) [°C.].

Whilst these equations might not be entirely representative for the conditions encountered with high speed engine (for example the area of the loops will probably depend upon gas density) they do indicate the dependence of α on p_m and c_m.

The considerable rise of the p_f values resulting from the rise of the pressure of the aspirated air, as shown in Fig. 5, is due to the consequent rise of mechanical losses, pumping losses and heat losses [as indicated by equations (4) and (5)] the latter increasing the areas of the loops Fig. 6. The more rapid rise of the losses in the region of higher speeds is due to the increased pumping losses, particularly with higher intake pressures, these also causing higher friction losses. So far as pumping losses are concerned it might be mentioned that the values of orifice coefficients for intake valves are about 0.8 to 1.2, depending on a number of design factors (7).

Similarly higher losses will be encountered at higher compression ratios (Fig. 7).

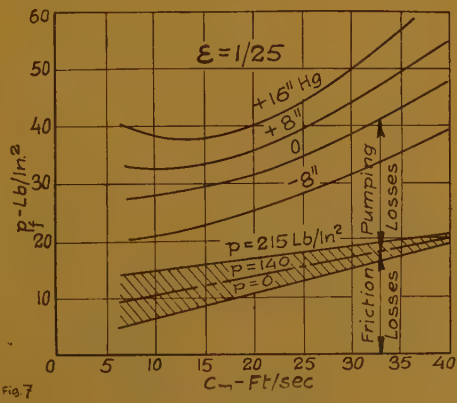


Fig. 7. — Effect of gas pressure on M.F.P. of engine.

The dependence of the losses upon the compression ration of an engine with a cylindrical combustion chamber are shown in Fig. 8. Here again the increase of the losses at low engine speeds is mainly due to heat transfer and leakage losses, whilst at higher speeds pumping losses predominate. The losses ascer-

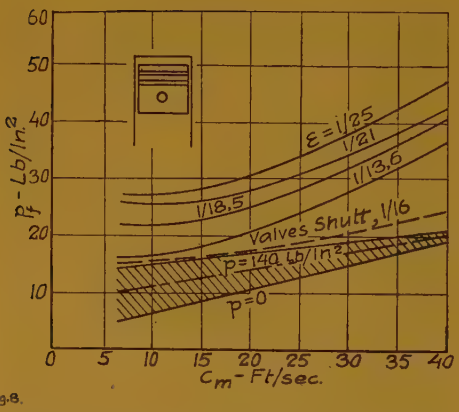


Fig. 8. — Effect of compression ratio on M.F.P. of engine with cylindrical combustion chamber.

tained with both the inlet and outlet valves closed are shown by the dotted line. These losses are caused by the leakage past the piston rings, both during the compression and expansion stroke.

The results of tests carried out with a turbulence chamber are shown in Fig. 9. As might be expected the losses are higher than in the case of the cylindrical combustion chamber Fig. 8, mainly because of the small clearance of 1 mm.

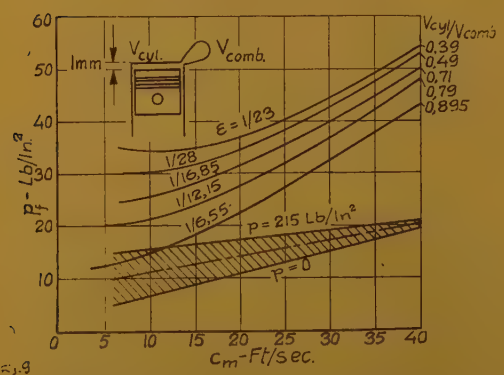


Fig. 9. — Effect of compression ratio on M.F.P. of engine with turbulence type combustion chamber.

With the piston approaching the top centre, high radial air velocities towards the turbulence chamber result in high hydraulic losses. An increase of the clearance from 1 mm. to 5 mm. resulted in an appreciable reduction of the losses at higher engine speeds as shown by Fig. 10.

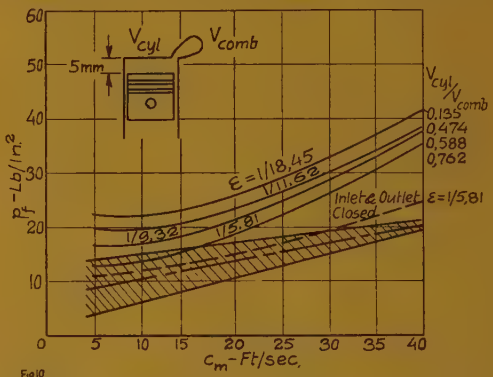


Fig. 10. — Effect of compression ratio on M.F.P. of engine with turbulence type combustion chamber.

So far we have only dealt with results obtained with motored engines, no restrictions being provided in the exhaust passages. If, however, the exhaust passage of an engine is throttled then the indicator diagram will be altered from that of Fig. 6 to the one shown on Fig. 11.

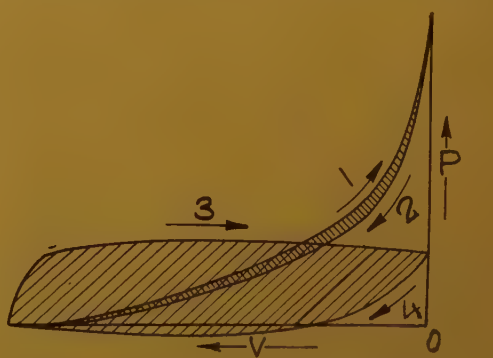


Fig. 11. — Indicator diagram of engine during braking.

After aspirating, compressing and expanding the air in the cylinder along the lines already indicated by the diagram Fig. 6, the exhaust valve will open and the air forced into the space between the exhaust valve and a throttle provided in the exhaust pipe. Upon completion of this stroke the exhaust valve will close, the inlet valve will open and the cycle will be repeated. The rapid pressure drop encountered in the cylinder at the beginning of the suction stroke 4 is caused by the air pressure between the piston and throttle, and later between piston and closed exhaust valve, dropping to the level of the aspirated air. This pressure drop is facilitated by the valve overlap, which with high speed traction engines amounts to 20° to 60°, with 25° to 30° as average values.

It will be realised that the amount of work done by the engine when driven by the vehicle will be also determined by the initial setting and rate of the exhaust valve springs. An unduly low setting would result in the valve admitting compressed air from the reservoir formed by the space between the exhaust valve and throttle during most of the suction stroke. The maximum compression pressure during stroke 3 of Fig. 11 will thus be determined mainly by the spring characteristics of the exhaust valve. The work required to compress one cubic foot of air at a pressure of p_0 to a pressure of p_m is given by the equation :

$$L = 200 V_1 p_m \text{ [ft./lb.]}$$

where :

V_1 = volume of aspirated air [ft³].

$p_m = b \cdot p_0$,

or :

$$L = 10\,000 V_1 p_m \text{ [m./kgr.]}.$$

where :

V_1 = volume of aspirated air [m³].

The values of b are plotted for compression exponent n varying between 1 (isothermal compression) and 1.4 (adiabatic compression) in Fig. 12. For the case considered here n can be taken as 1.35.

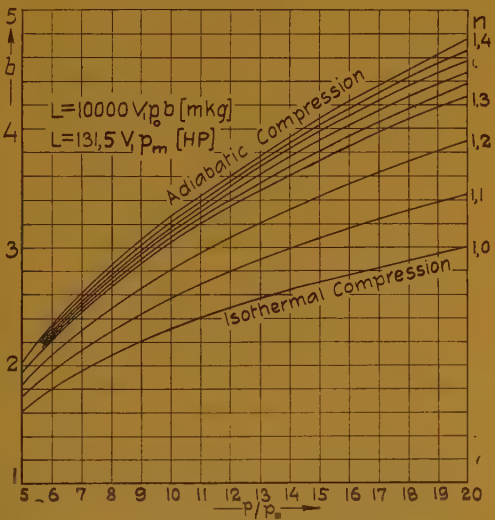


Fig. 12. — Factor 6 as a function of pressure ratio and polytropic exponent.

It is clear that because of the great valve overlap (up to 150°) normally provided with supercharged engines to ensure efficient scavenging, the brake power developed during stroke 3 and 4 will be appreciably smaller.

The effect of exhaust back pressure upon losses of a single-cylinder unit are shown in Fig. 13.

The data dealt with so far applied to single-cylinder units only. The losses due to big end bearings and piston friction will be similar for multi-cylinder engines, whilst the crankshaft bearing losses will be smaller. The general dependencies will be very similar and because of this the data presented above

may be considered as generally indicative of what may be expected with a multi-cylinder engine utilised as vehicle braking medium. For example the brake power/power ratio of a 4.85 litre six

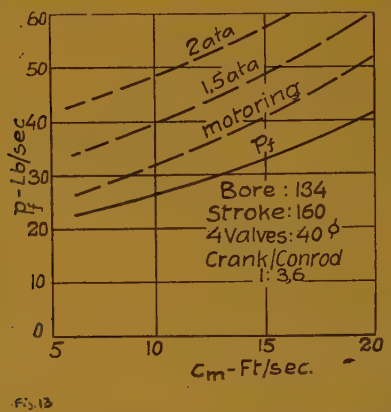


Fig. 13. — Effect of exhaust back pressure on M.F.P.

cylinder engine running with a free exhaust is shown in Fig. 14 (8). This merely confirms the data shown in Fig. 5, 7 to 10 and 12 for since with the mean effective pressures of modern traction diesel engines at top speed is about 85 to 100 lbs. per sq. inch, the brake power/power ratio with unthrottled exhaust would be about 50 %. To this should be added the power required to drive the

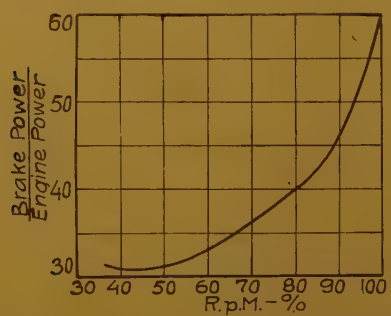


Fig. 14. — Average ratios of M.F.P./M.E.P. for diesel engines with open exhaust.

auxiliaries such as cooling fan, coolant pumps, lighting dynamos and brake compressor as well as the « lost power » absorbed by the air cleaning and exhaust systems.

The possibilities offered by the use of the exhaust brake with railcars provided with a mechanical transmission will be gathered from the following example. Consider a 25 ton articulated type railcar powered by a 200 H.P. engine driving one axle via a four speed transmission. The fan absorbs 5 H.P. at maximum engine speed, whilst the exhaust and air cleaners absorb 3 and 4 H.P. respectively The power absorbed by the former is proportional to the cube of the speed, that absorbed by the latter two is proportional to the square of the speed. The power required to drive the dynamo is assumed to 1.5 H.P. The over-all transmission efficiency is assumed to 90 % over the entire speed range. The drive is to one axle carrying a load of 7.6 tons.

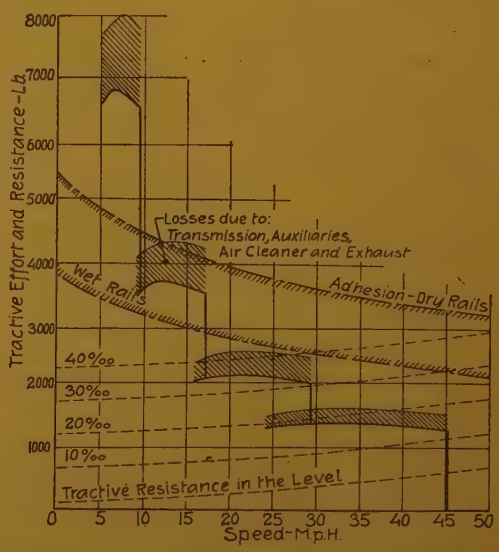


Fig. 15. — Tractive effort and resistance curves of a 25 t., 200 H.P. railcar.

The relevant tractive effort values in various gears are shown in Fig. 15 together with the maximum tractive effort as limited by adhesion and tractive resistance values for a number of gradients. The grades which can be climbed in various gears are shown in Fig. 16.

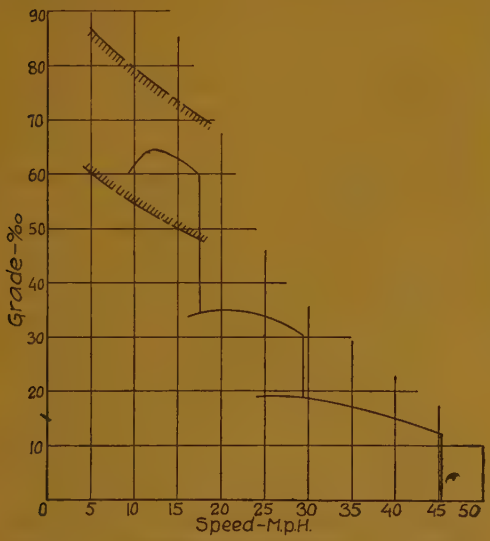


Fig. 16. — Gradient performance of railcar.

Similar data for the braking performance is given in Figs. 17 and 18. In either case the tractive resistance is ascertained in accordance with the equation :

$$R = 4.5W + 0.525cF\left(\frac{V + 10}{10}\right)^2 \text{[lb.]} \quad (7)$$

where :

W = vehicle weight [ton].
F = frontal area [ft²].
V = velocity [miles/hr.].
c = drag coefficient (for car with rounded front considered here $c = 0.3$).

The values for the coefficient of adhesion during braking are plotted in accordance with test results obtained by METZKOW (9) and also by the *Chicago, Burlington and Quincy Railroad* (10).

A comparison of Figs. 16 and 18 shows

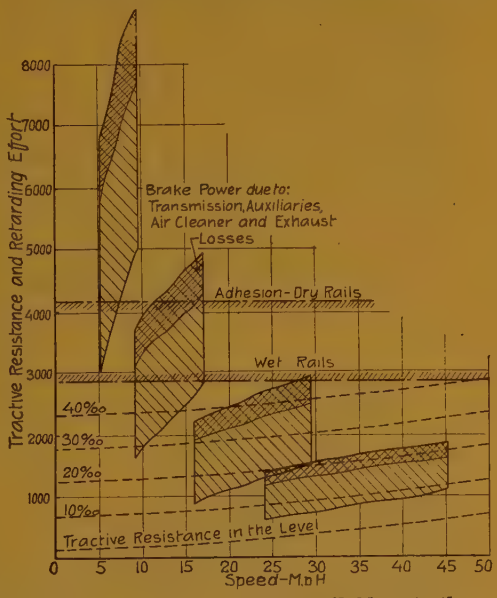


Fig. 17. — Brake power available at the wheels.

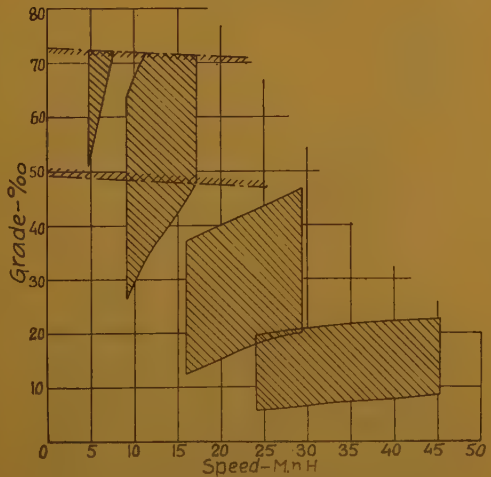


Fig. 18. — Speed down grades with engine acting as brake.

that the vehicle will be capable of climbing and descending approximately the same grades at similar speeds with the engine driving or acting as a brake, respectively. This has been confirmed by experience with road vehicles.

In addition the exhaust brake can be also employed for retardation. The deceleration obtainable with the 200 H.P. engine used in the 25-ton vehicle dealt with here is shown in Fig. 19, the speed-time and speed-distance curves being plotted from the data of Fig. 17.

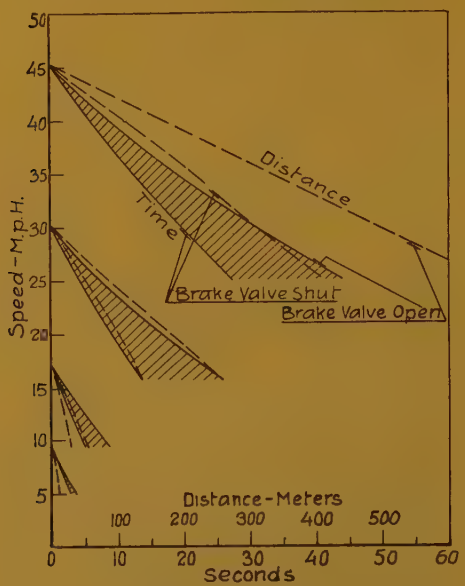


Fig. 19. — Retardition of car with engine acting as brake.

The use of the exhaust brake as a means of ensuring a constant vehicle speed when running downhill is extremely popular in Switzerland where it is being widely used with both lorries and buses in order to converse brake linings. The operating results obtained with the exhaust brakes fitted to road vehicles operating both city and interurban services are extremely satisfactory and because of this exhaust brakes are now regarded as standard equipment throughout the country. The action of the exhaust brakes is very smooth, gradual and powerful and can be easily controlled by means of a hand lever usually provided at the driver's right and operated in the horizontal plane.

Whilst in the early days of its development the use of exhaust brakes lead to undue complications (For example engines were fitted with axialy movable camshafts provided with an additional cam. Exhaust braking was achieved by keeping the inlet valve closed) at present efficient braking is ensured merely by the provision of a throttle in the exhaust. In the case of vehicles manufactured by the Saurer Company of Arbon, Switzerland, this valve is a simple butterfly unit provided with suitable heat resisting glands. A valve developed by Oetiker of Zurich is of a somewhat more complicated design. Since this valve (Fig. 20) is of the sliding type, provision must be made for easy operation particularly when approaching the fully closed position, since here the gas pressure would press the valve onto the seat and thus make operation difficult. To prevent this a secondary, unloading opening 2 has been provided in addition to the main port 1. The valves 3 and 4 are linked in such a way that when closing valve 3 covers port 1 before valve 4 covers port 2, whilst when opening valve 4 is actuated first thus relieving valve 3 of the load due to gas pressure. An aperture 5 is provided for connection to an air reservoir which might be charged by the engine via a non-return valve. This feature is particularly attractive for use in mountainous service where the engine

can thus be called on to help in charging the air brake reservoirs.

The weight of the brake valves is relatively low and because of this they can be supported directly by the pipes, being mounted fairly close to the engine. In the case of supercharged engines the valve should be arranged between engine and supercharger — for obvious reasons.

Whatever the brake valve it must be interconnected with the fuel control so as to prevent the possibility of operating the valve with the fuel supply on. A simple way of ensuring the requisite interlocking — due to the Saurer Company — is shown in Fig. 21. The lever operating the brake valve also shuts off the fuel supply. With the brake valve shut, operation of the throttle lever will merely release the spring S. A return of the brake valve into the open position will automatically restore the fuel supply to the engine, provided the position of the throttle control lever demands this. This means, of course, that the exhaust brake must be released shortly before bringing the vehicle to a stop, if the engine is to remain running.

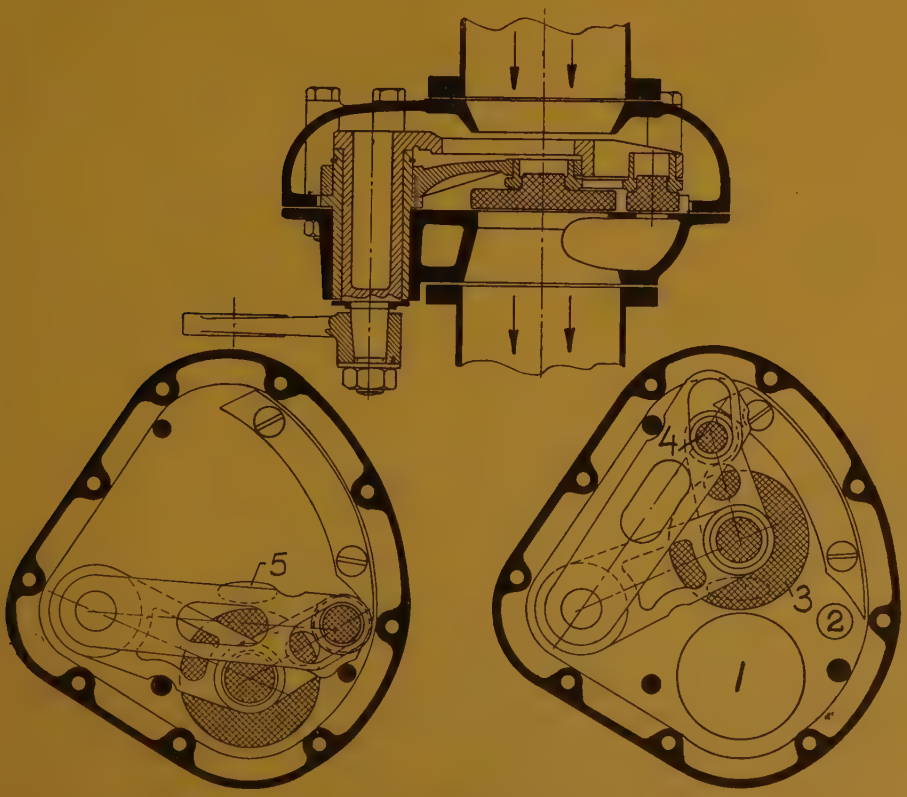


Fig. 20. — Oetiker brake valve.

The possibilities of overspeeding the engine by selecting a wrong gear when applying the exhaust brake, needs of course to be impressed on drivers, but no undue difficulties should be experienced in this respect.

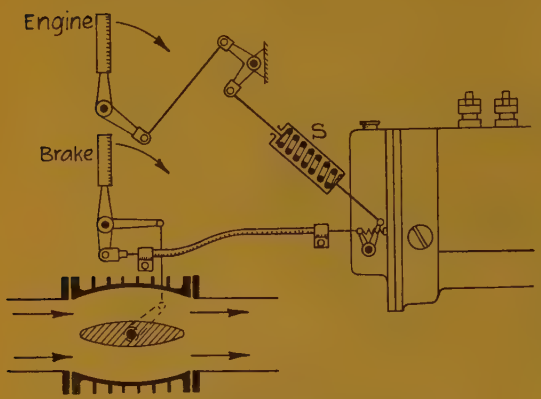


Fig. 21. — Throttle and brake control.

Needless to say exhaust braking can be employed not only with vehicles provided with mechanical transmission but when using electrical or hydraulic transmissions as well.

The author wishes to acknowledge his indebtness to Mr. V. DIEM, Director of the S.A. Adolphe Saurer of Arbon for the opportunity given to him for studying the operation of the exhaust brake in service.

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The future railcars of the French National Railways,

by TOURNEUR,

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(Revue Générale des Chemins de fer, March 1947.)

As a result of the war and looting by the enemy the present stock of railcars belonging to the French National Railways is only about 75 % of that of 1939. Amongst those that remain are many that were seriously damaged, and in spite of the repairs made to them, their life is not likely to be long; in fact about the half of the stock is more than ten years old, while it seems that the life of a railcar should not exceed fifteen years. The French National Railways have therefore been forced to make provision for building some 400 railcars and 500 trailers over the next ten years.

In the present note, we propose to give the general characteristics of the types retained, for reasons made clear in a preliminary report of the conditions under which these railcars will be used. The policy of the French National Railways in this respect is obviously based on the experience acquired by the former Companies and subsequently its own, so that we will begin by a short historical review of the evolution of railcars since 1931, when the first trials were carried out.

I. Evolution of railcars before the war.

a) 1931-1934 period. — In 1931 and 1932 the French railways, in an endeavour to fight road competition, decided to carry out experiments with light railcars, designed above all for services on the secondary lines and for stopping or semi-through trains, profiting by the possibilities of weight reduction offered by the recent introduction of high speed diesel engines and the newest designs of all metal bodies. As a result the first « railcars » were put into service in 1933 and 1934, a name which may be criticised but which has the merit of calling to mind their analogies with road vehicles : lightness and high power weight ratio.

For example the first « Michelines » were put into service in 1932 by the Michelin Company which has to its credit many years of research into the possibility of adapting pneumatic tyres to railway traction.

The railcars of this period were mostly of small capacity (about 45 seats) equipped with the same engines as road vehicles, the nominal horse power of which varied between 80 and 135, but was considerably lower in actual fact. Some typical railcars of the period are shown in the table on page 574 (Nos. 1 to 4).

These original railcars had many technical defects, and though they were used for stopping services on secondary lines, their effective power weight ratio was often too low to enable the relatively high average speeds then required to be attained; moreover the co-ordination measures in force at the time led to most of the lines for which they were intended being closed down. As a result in the years that followed the companies gave up designing small low power railcars.

At about the same period larger railcars equipped with more powerful diesel engines, specially designed for the railway, were given a trial (the most typical example of this category is the

Renault type VH railcar, with two bogies, weighing 23 t. empty, with 78 seats, and a 220 H.P. engine). These railcars, which could run at 110 km. (68 miles)/h. and had very good acceleration (Nos. 5 to 7 in the Table) could be used not only for stopping services but also for through and semi-through services.

The average speed was about 90 km. (55 miles)/h. on the through services, and 45 km. (28 miles)/h. on the stopping services; the latter figure in particular was appreciably higher than that obtained with steam traction.

Owing to the particular importance then attached to maintaining such average speeds, on account of the relatively low power of the engines then available, it was necessary to use particularly light bodies and impossible to use any trailers. Used in this way however railcars were incapable of meeting peaks of traffic, since the capacity offered was practically invariable; it was consequently necessary to multiply the services, which was not always possible, or else to run steam trains in their place on days when the traffic was heavy, usually an expensive solution.

	TYPE	Year of construction	Nominal power of engines	Maximum speed	No. of seats	Useful area	Weight empty per m^2 of area	Power per ton of load
			H.P.	km/h		m^2	t/m^2	H.P./t
1	*SOMUA 4-wheeled*	1933	80	90	50	23.33	0.48	4.8
2	*Charentaises with 4 parallel pairs of wheels* (Fig. 1).....	1933	89	90	62	32.50	0.31	4
3	*Charentaises 4-wheeled* .	1934	100	90	55	25	0.30	5.3
4	*Renault ZO*	1934	110	90	35	21.50	0.43	6.6
5	*Renault VH type* 1932..	1933	220	110	78	38	0.30	7
6	*Renault ABJ* (Fig 2) ...	1935	300	120	90	48	0.64	7.6
7	*De Dietrich* 320 *H. P.*.	1936	2×160	120	90	49	0.71	7.4
8	*Acieries du Nord (Standard)* (Fig. 3).....	1933	2×300	120	88	44	1.07	9.3
9	*Renault ADX2*	1939	2×300	126	88	46	0.91	11.8
10	*Decauville* 640 *H. P. (electric drive)*	1939	2×320	110	89	46.50	0.90	13
11	*Renault ADP*	1938	500	125	84	45.50	0.88	10.9
12	*Micheline* 56 *seats*	1934	230	110	72	37	0.27	12.8
13	*Micheline* 96 *seats*	1937	420	120	114	66	0.33	12.8
14	*Bugatti* 400 *H. P.*.....	1935	2×200	140	68	54	0.59	10.8

b) 1935-1939 period. — During this period, for the reason mentioned above, the need for constantly increasing the capacity of railcars was kept in mind in all new designs :

— by an increase in the length of the body, which was 30 m. (98'5½") on the latest *Michelines;* thanks to improvements made in the diesel engines, their power was increased to 265/280 H.P.

and subsequently 300/320 H.P. so that the power weight ratio remained unaltered;

— the use of doubling, consisting of coupling two railcars together, each driven by its own driver, the leading driver transmitting to the second one the necessary information by means of optical or acoustic signals;

Fig. 1.

— the designing of railcars which could run coupled together, the rear unit being controlled from a distance by the driver of the leading car;

— the construction of articulated units, with two bodies carried on three bogies, with two 300 H.P. engines;

— finally, after 1937, the putting into service of appreciably more powerful railcars, with two 300/320 H.P. engines or a single 500 H.P. engine, capable of hauling one or two light trailers, or an ordinary coach (Nos. 8 to 11 of the table).

This brief historical review would be incomplete if we omitted to mention two

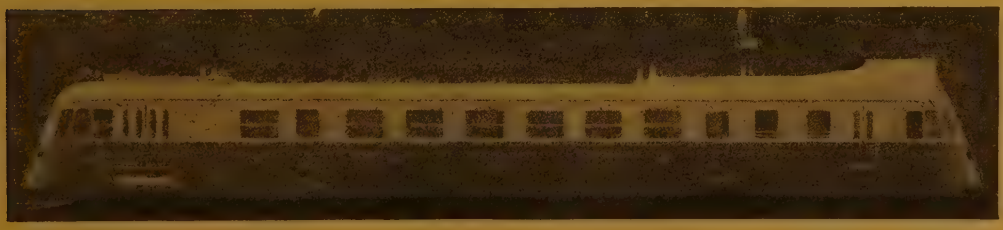


Fig. 2. — Renault railcar type ABJ.

Fig. 3. — Standard Nord railcar with trailer.

special categories of railcars, the *Michelines* and the *Bugatti* railcars, both of which were equipped not with diesel engines but with petrol engines (Nos. 12 to 14 of the table).

Between 1932 and 1939 the *Michelin* Rubber Company supplied the French railways with a hundred light railcars,

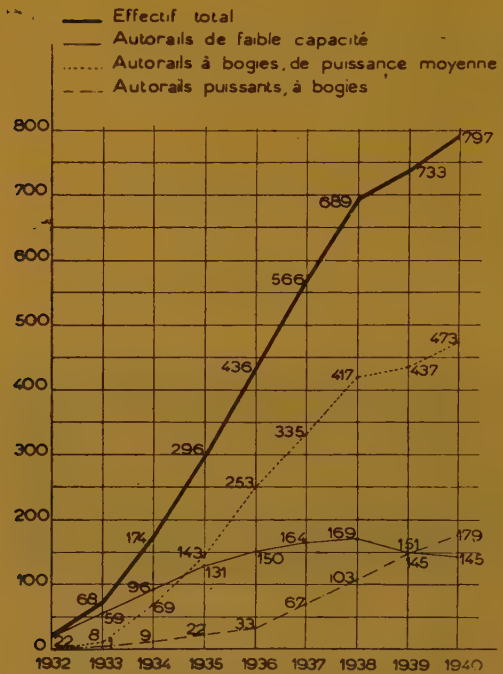


Fig. 4. — Graph showing the French National Railways stock of railcars.

Explanation of French terms:

Effectif total = Total strength.

Autorails de faible capacité = Railcars of small capacity.

Autorails à bogies de puissance moyenne = Bogie railcars of medium power.

Autorails puissants à bogies = Powerful bogie railcars.

on pneumatics, with petrol engines, the prototypes of which were tried out in 1930. This firm played a large part in the development of railcars and contributed largely to their success in particular by their trials into weight reduction.

The *Bugatti* railcars, which had many original features, were specially designed for high speed through runs on the main lines. In this special field, they were used at the same time as the diesel units supplied by the *Franco-Belge* Company to the Nord railways.

At the beginning of the war, the French National Railways therefore owned some 800 railcars, three-fifths of which were 300 H.P. bogie units designed to run singly or in pairs, but not suitable for trailers; the remainder consisted of about 150 low power units, the stock of which had remained unaltered since 1936, and about 180 powerful units, capable of hauling trailers, mostly built since 1936 (Fig. 4).

In order to build up its stock, reduced to 600 owing to the war, the French National Railways have made provision in their ten-year programme for the construction of :

— 400 railcars of three different types, to which a few small railcars will be added for small lines, and

— 500 special trailers.

We will now describe the leading characteristics of these vehicles :

II. Conditions under which the future railcars will be used. Choice of types to be adopted.

Before the war railcars were used, as we have already explained, for very diverse services from express runs over long distances *(Paris-Lille, Paris-Strasbourg, Paris-Lyons, etc.)* to the services on the few small secondary lines unaffected by the co-ordination measures; express parcels traffic was even worked by means of railcar-vans. But although the railcar was capable technically of adaptation to all these different services, giving in particular the additional speed

required, in practice it was found necessary to limit its field of action.

In the case of the ultra-fast services, the capacity of the railcars was soon found to be insufficient owing to the great favour such services found amongst passengers, so that in the future the French National Railways propose to cover these by means of real trains, the method of traction used varying according to the line; it has already been arranged that the *Paris-Lille* and *Paris-Le Havre* services will be covered by multiple diesel units. Such units cannot be classified as railcars owing to their large size, so we will not deal with them here.

As regards railcars properly so-called, we have seen that the chief difficulty encountered in their use lies in the fact that it is difficult to provide for peak periods. To improve matters from this point of view it is absolutely essential to have the engines sufficiently powerful to enable a trailer to be added when the traffic makes this necessary without appreciably lowering the average speed.

As soon as the fact that it runs on its own is no longer considered to be an essential characteristic of the railcar, it becomes necessary to define the limit of its power; owing to the increase in the power of fast diesel engines, it will soon be possible to design twin-engined units with a power of more than 1 000 H.P. with which, in spite of this, part of the body will be available for use by passengers or their luggage in order to obtain the necessary minimum adhesion; it would be out of place to call such units « railcars », however, as they can be used in exactly the same way as a locomotive. It seems to us that the power of a railcar should be limited to 600 H.P., as if this power is exceeded it is not possible to run the railcars as single units, even on the most difficult sections of line, at a reasonable cost. On the other hand, a 600 H.P. railcar can haul a light bogie trailer on most of the very hilly lines of the French National Railways, and can haul about 100 tons on the easy sections; this power therefore gives it all the flexibility in working that is needed.

Another source of difficulty encountered in the use of railcars lies in the great diversity of types in service which in practice makes it necessary to group together at the same centre those which are the most alike, in order to facilitate the work of the driving and maintenance staff; this entails some drawbacks from the operating point of view, and means that the railcars available are not always those best suited to the needs of the service in question. Consequently in the new programme it was essential to impose a strict limit on the number of types and to make their components as standardised as possible.

The types of railcars retained by the French National Railways come under two categories :

— the first includes railcars of relatively small capacity, with a moderate power weight ratio, but capable of hauling a light trailer, which are designed above all for serving secondary lines economically;

— the second consists of bogie railcars, with a higher power weight ratio, capable of running at appreciably higher speeds than the steam trains, even when hauling trailers.

These different railcars are required to meet the following requirements :

— run through services on cross country lines between main lines where there are no convenient train services;

— on certain important arteries where the stopping trains do not carry much traffic, complete the express services by collecting or distribution services connecting with the express trains at the important stations;

— give a third service (in the middle of the day) on lines which are already

served by means of steam stopping trains morning and night;

— replace steam trains on lines with little or very little traffic.

a) Railcars belonging to the first category (150 H.P. at the most). — The capacity of railcars of this type has been determined in such a way that the desired performance can be obtained with the largest engines in use on road vehicles in order to profit by the saving in cost due to mass production. This consideration led to the 150 H.P. engine being adopted, so that the weight has been limited to 30 tons loaded, to which a light trailer can be added.

There is room for 80 passengers in the railcar (Fig. 5) (60 seats), together with a luggage compartment, and it can haul a light four wheeled trailer that will carry 60 passengers; its maximum speed will be about 100 km. (62 miles)/h. by itself, and 80 km. (50 miles)/h. with a trailer.

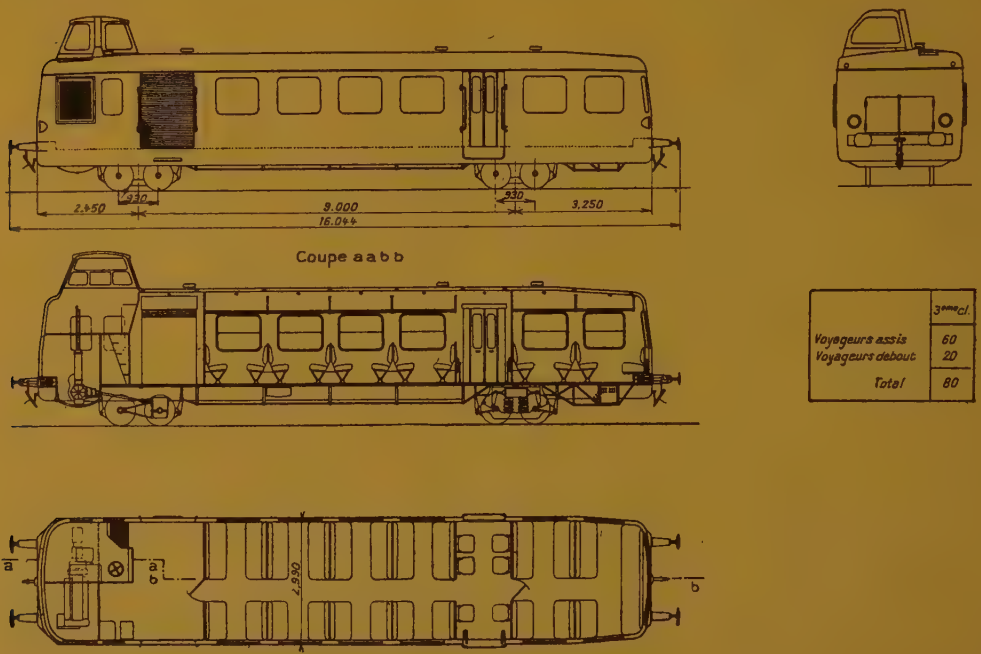


Fig. 5. — Diagram of the 150 H.P. railcars.

Explanation of French terms:

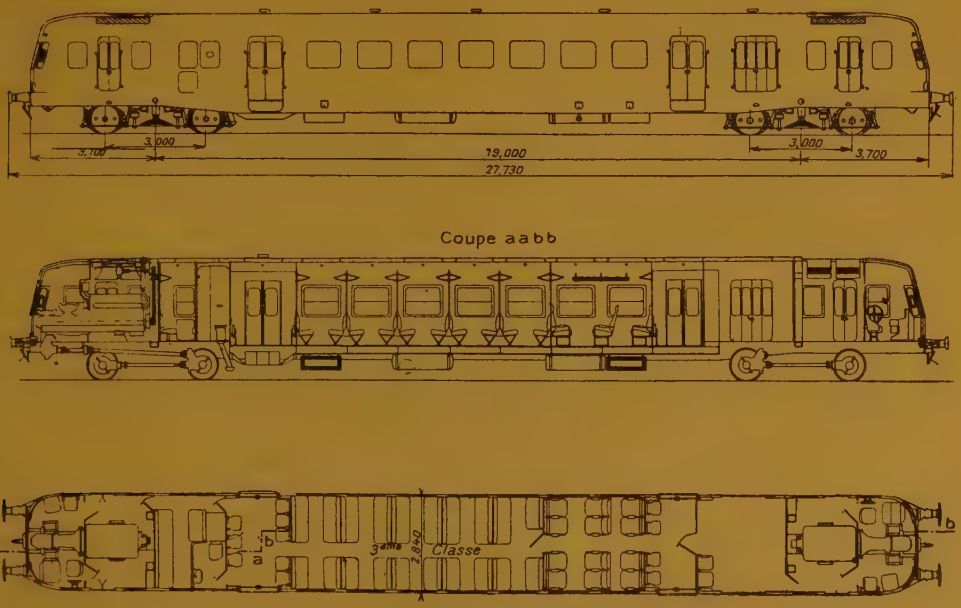
3e cl. = 3rd class.
Coupe = Section.
Voyageurs assis = Seats.
Voyageurs debout = Standing room for.

It will be seen that the fact that a 150 H.P. engine is being used, taking into account the technical progress made, has made it possible to get a vehicle with an appreciably higher transport capacity than the first railcars built about 1934, especially when the fact that a trailer can be added is taken into account. The general arrangements

adopted, to which we will return later, should nevertheless lead to a moderate kilometric cost .

But even taking into account the traffic allocated to the road in the case of secondary lines, the French National

It is proposed to work such services either by means of small four wheeled railcars or by adapting road vehicles to run on the railway. This latter proposal which was tried once before without much success now seems to have



	2nd. class.	3rd. class.	Total.
Seats	12	56	68
Standing room for	12	30	42
Total	24	86	110

Fig. 6. — Diagram of the 600 H.P. railcars.

Railways still have to assure passenger services on a certain number of small lines where the traffic is insufficient to justify the use of even the above type of railcar, so that in such cases it may be an advantage to reduce the capacity and increase the number of services.

possibilities thanks to the increased strength of the bodies of standard road vehicles built according to principles followed for the last twenty years in the case of railway rolling stock (monoblock steel construction in which the body sides take part of the load) as well

as improved springing and the perfecting of wheel centres with tyres carried on springs.

Owing to the small running resistance offered by the assembly « metal wheel-rail » compared with a pneumatic tyre on the road, a motorbus altered in this way, hauling a trailer which doubles its capacity, will run at a higher speed than it is capable of on the road by itself, the fuel consumption being the same in both cases.

Operating trials with motorbuses equipped in this way are in hand.

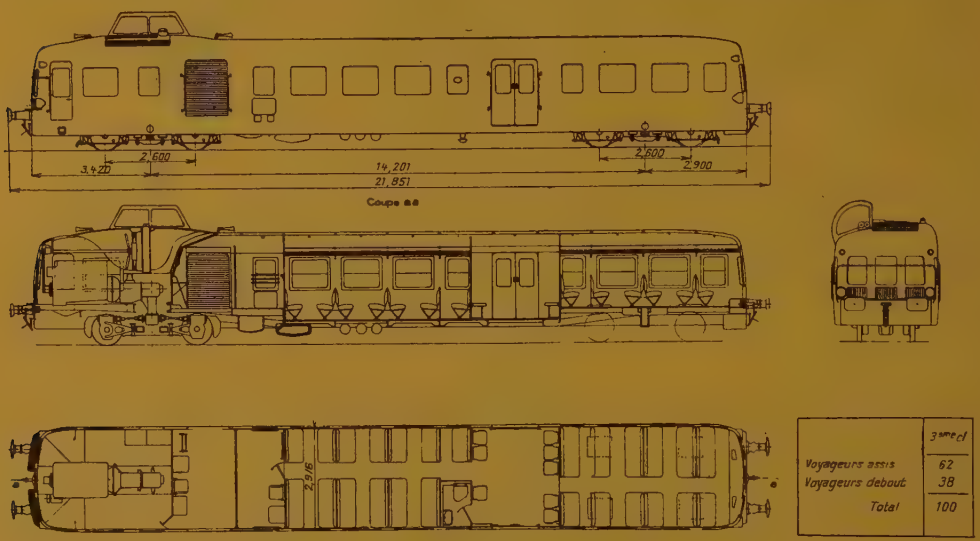


Fig. 7. — Diagram of the 300 H.P. railcar.

Explanation of French terms:

Coupe = Section.
Voyageurs assis = Seats.
Voyageurs debout = Standing room for.

b) Railcars of the 2nd category (300 to 600 H.P.). — In this category the French National Railways are building twin engined railcars (2 × 300 H.P.) (Fig. 6), derived from similar types now in service which have given particular satisfaction. Such railcars, however, the power of which is exactly equivalent to that which we gave above as the maximum desirable, cannot, for the following reasons, cover economically sufficient of the services for which the railcars in the first category are inadequate:

— running a single 600 H.P. railcar can only be justified on very difficult sections of line, and then only when an exacting timetable has to be worked; many services on easy sections or where the gradients are only slight require a less powerful engine, yet the railcar in question must have a capacity, performance and comfort at high speeds exceeding those possible with the 150 H.P. railcars;

— in the present state of technical knowledge, the 600 H.P. railcar is necessarily twin-engined, which increases the horse power cost and involves complicated maintenance owing to the greater complication of the equipment. The

services on many of the secondary lines however involve the use of country railcars overhaul of which, apart from the daily check by the driver, must be at as long intervals as possible. It may be argued, it is true, that with the twin-engined railcars breakdowns are practically unknown; but in view of the safe running of present day engines, this argument does not carry weight in our opinion except on very extensive systems with very light traffic, which is not the case with the French lines.

A second type of railcar, equipped with a single 300 H.P. engine (Fig. 7) has in consequence also been retained; it can run coupled up to a second railcar of the same type, and a trailer can be added to them when the profile of the line so allows.

The diagram (Fig. 8) shows all the possibilities of the 600 and 300 H.P. railcars. It shows that certain compositions lead to comparable capacities and performances; the cost per mile of these compositions are also very similar. This must not lead to the conclusion that the 600 H.P. railcar is superflous : apart from its high power weight ratio which makes it possible for it under good conditions to haul ordinary rolling stock (a bogie coach or parcels van for example), it is better suited than the 300 H.P. railcar in mountainous regions to an increase in the power weight ratio of the train on difficult sections, to the detriment of its capacity be it clearly understood. In general therefore it will be necessary to have the three types of railcars at the depots, the 600 H.P. type being the predominant one in the mountainous districts (Massif Central and Alps).

Retaining these three types of railcars

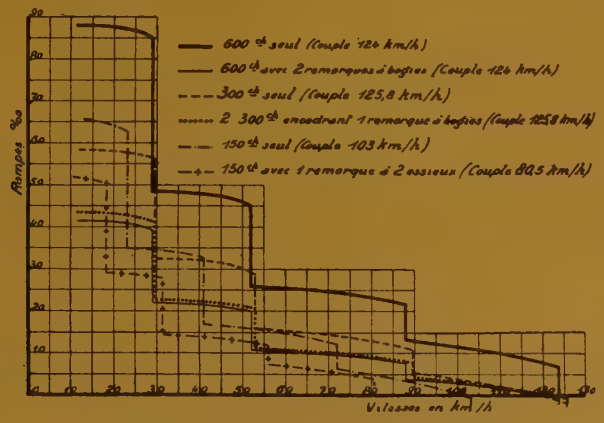

600 ch seul (couple 124 km/h)
600 ch avec 2 remorques à bogies (couple 124 km/h)
300 ch seul (couple 125,8 km/h)
2 300 ch encadrant 1 remorque à bogies (couple 125,8 km/h)
150 ch seul (couple 103 km/h)
150 ch avec 1 remorque à 2 essieux (couple 80,5 km/h)
Rampes ‰
Vitesses en km/h

Fig. 8. — Performances of future railcars.

Explanation of French terms :

600 H.P. alone (torque 124 km./h.).
600 H.P. with 2 bogie trailers (torque 124 km./h.).
300 H.P. alone (torque 125.8 km./h.).
two 300 H.P. with a bogie trailer (torque 125.8 km./h.).
150 H.P. alone (torque 103 km./h.).
150 H.P. with 4-wheeled trailer (torque 80.5 km./h.).
Rampes = Gradients.
Vitesses en km./h. = Speed in km./h.

will necessitate the use of two kinds of trailers : one a four wheeler intended for use with the 150 H.P. railcars in service on secondary lines, the other a bogie trailer (Fig. 9) of large capacity, which will be used with railcars of the second category.

vice. It is not intended to use pre-heaters using petrol, as the saving in maintenance costs have not been found sufficient to make good the extra cost of fuel involved (*).

On the types of railcars adopted, the engines will be fitted in the body and

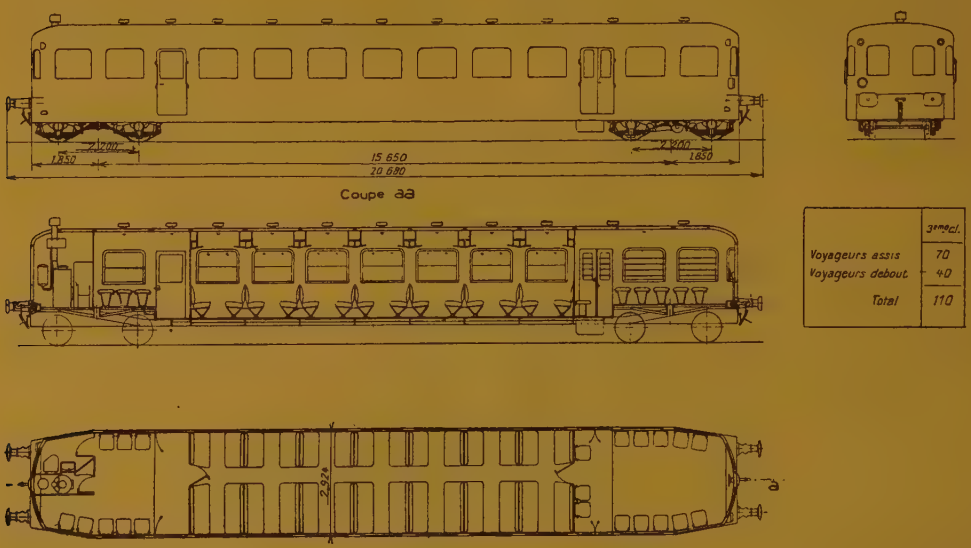


Fig. 9. — Diagram of the bogie trailer.

Explanation of French terms:

Coupe = Section.
Voyageurs assis = Seats.
Voyageurs debout = Standing room for.

III. The principal technical characteristics of the types of railcars and trailers adopted.

Detailed descriptions of the three types of railcars will be published when these are put into service. We will limit ourselves therefore in this note to pointing out the essential characteristics of this stock.

The engines. — In principle all the new railcars will be equipped with high speed diesel engines (1 500 revs per min.) with mechanical injection, using gasoil, of types already tried out in service. It is not intended to use pre-heaters using petrol, as the saving in maintenance costs have not been found sufficient to make good the extra cost of fuel involved (*).

On the types of railcars adopted, the engines will be fitted in the body and not on the bogies, so as to lighten the latter to the maximum possible extent and thus improve their running.

(*) This will involve giving up the construction of pneumatic tyred railcars which require particularly light engines. It would however be a pity not to profit by the comfort which pneumatic tyres give to railway rolling stock, consisting above all in the almost complete suppression of noise and vibration. For this reason the French National Railways, profiting by the experience already obtained with their railcars, are having three pneumatic tyred sets built under the supervision of the *Micheline* Company for the fast day services between Paris and Strasbourg.

The transmision. — Although the working of the railcars with electric transmission now in service has been satisfactory, the future railcars will have mechanical transmission.

In the case of the 150 and 300 H.P.

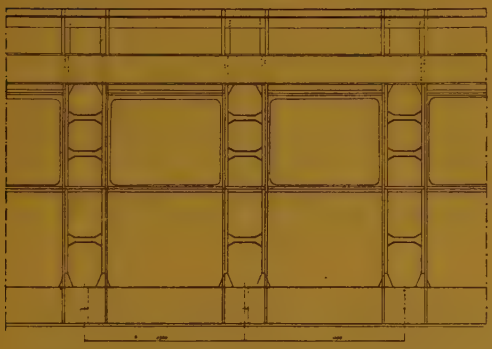


Fig. 10. — Detail of frame of 600 H.P. railcar.

units, they are very satisfactory, and cost a little less than electric transmission. Their chief advantage is the fact that they make it possible for the full power of the engine to be used at very different speeds (in practice lying between 30 and 120 km. [18 and 75 miles]/h.), so that the same railcar can be used for very different services, from fast runs on level lines to stopping services on mountain lines, with a trailer. This advantage is further increased by the fact that to meet present day requirements many railcars with mechanical transmission designed to run as single units at high average speeds have been adapted for slower services where they haul trailers; the same cannot be said for railcars with electric transmission.

The first series of railcars to be built will be fitted with *Renault* 150 and 300 H.P. transmissions of types already tried out on many cars already in service.

It would moreover seem to be possible, in view of the progress made in this field, to fit a new 450 H.P. transmission in the space taken by a 300 H.P. transmission, and as this power will shortly be obtained by supercharging the present 300 H.P. engines, the power of the 300 and 600 H.P. railcars could easily be increased by nearly 50 % if necessary.

The bodies. — The bodies of the 600 and 300 H.P. railcars will be made of steel, with the sides taking part of the load, of all welded construction; the frame is of the Vierendeel type without diagonals (Fig. 10). This method of construction ensures great rigidity, which is particularly vital on account of the large distance between bogie centres.

The body of the 150 H.P. railcars, the details of which have been designed by the Renault Company, is very similar to the bodies of many railcars already built by this firm : steel framework with diagonal bracing in the panels to which the internal and external light metal sheets are fixed; all the parts of the frame are made of square tubing (Fig. 11). This method of construction

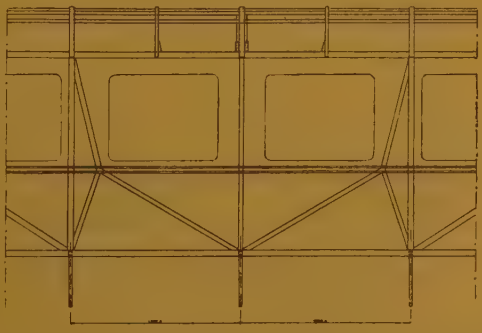


Fig. 11. — Detail of frame of 150 H.P. railcar.

makes it possible for most repairs to be carried out by the depots which is a very great advantage for these railcars which are used on lines at a distance from the large centres.

On all future railcars specially designed fireproof partitions will separate the

driving compartment from that occupied by passengers.

Finally in order to lower the cost and simplify the maintenance of the 150 and 300 H.P. railcars, these vehicles will have a single driving compartment near the engine. This arrangement, already tried out on other railcars, makes it possible to simplify the drive, make use of direct control, which is certain in working, while the engine is always in full view of the driver. The driving compartment, arranged on one side, and the roof of the railcar have been specially designed to give satisfactory visibility in either direction of running.

Bogies, running gear, suspension. — The 600 and 300 H.P. railcars have two light weight bogies, the framework of which, welded throughout, consists of relatively thin sheets.

On the other hand, in order to reduce the weight and lower the cost price of the 150 H.P. railcars, which will run at moderate speeds, four parallel axles have been provided, in two groups of two axles, similar in arrangement to that adopted in 1934 on the light railcars built by the Entreprises Industrielles Charentaises.

On all three types of railcars, there are roller-bearing axle boxes and one piece rolled steel wheels in special steel held by a nut on a taper wheel seat,

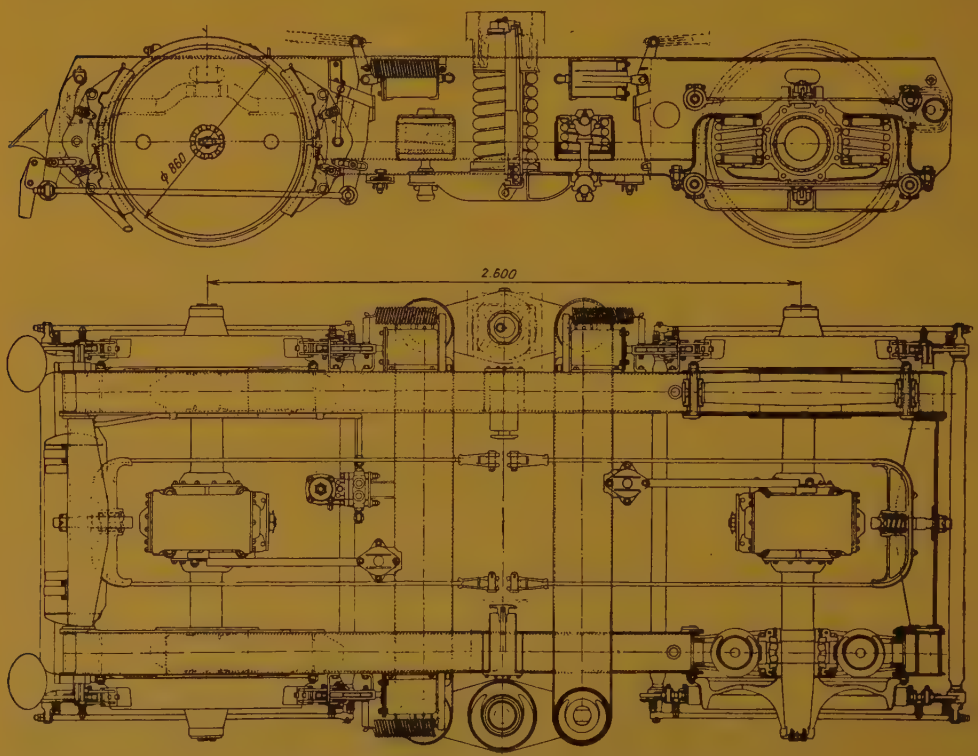


Fig. 12. — Bogie of the 300 H.P. French National Railways railcar.

which makes it easy to inspect the boxes and wheel seats. The brake rigging is easily reached, outside the main frame. Profiting by experience acquired with railcars in service, coiled springs have been used above all in the suspension, together with hydraulic shock absorbers.

Finally everything possible has been done to make greasing unnecessary, and prevent all play which is very prejudicial to good running, by avoiding parts with rubbing surfaces. In the bogies (Fig. 12) axle box guides have been

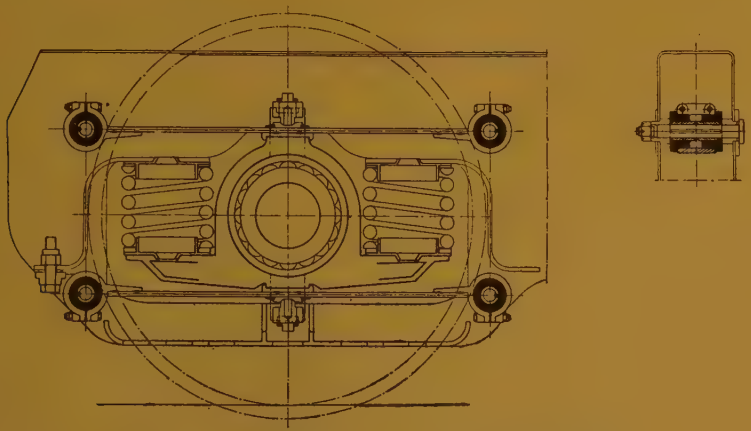


Fig. 13. — Diagram of the bogie suspension.

suppressed and the horizontal play between the frame and the axle boxes is transmitted by two leaf springs fixed respectively to the top and bottom of the box in the middle, the outer eyes being fastened to the body by links fitted with moulded rubber sockets; the primary suspension consists of coiled springs.

In the case of the motor bogie of the 300 H.P. railcar (Fig. 13) the mechanical transmission has been brought down through the centre and there is no king pin; traction and braking efforts are transmitted to the body by cables.

The brakes. — The railcars in service are equipped with either ordinary brakes with shoes, or with drum brakes.

It has been found in practice that the latter must be fitted on the outside of the wheels to give easy access to the fittings; but in this case the gauge limits the size of the drums very closely, so that, taking into account the need to limit the specific pressure of the fittings to values compatible with reasonable wear, it is not possible to adopt this method of braking on pairs of wheels with a load of more than 8 t.

Only the 150 H.P. railcars can therefore be fitted with this type of brake; it has been found, however, that well designed brake shoes will be cheaper, easier to maintains, and practically as light.

The three types of railcars and their trailers are therefore equipped with the classic type of brake, but with double shoes in order to reduce the specific pressure and so improve the performance whilst reducing wear.

The brakes are operated by compressed air with JMD equipment assuring at one and the same time automatic working, and moderate pressure and release,

and making it possible to use as a trailer an ordinary vehicle equipped with Westinghouse brakes. On the 150 H.P. railcars and their trailers, with low tares, compared with the useful load, braking when loaded is realised by modifying the adjustment of the regulator and loosening the suspension springs.

Cooling the engines, heating. — On the 150 and 300 H.P. railcars the driver can more or less regulate the water cooling system by regulating the amount of air admitted to the radiator. On the 600 H.P. railcars the engine water circuit includes an automatic by-pass of the radiator controlled by thermostat, which keeps the temperature of the water between 60° and 70° C. (140 and 158° F.).

On all these railcars which usually have to stand out in the open all the cooling water can be collected in an insulated tank under the floor, in which it remains warm during the night; when it is very cold a small petrol heater keeps it warm.

The compartments are heated by aerotherms under the seats branching from the radiator of the diesel engine; the electric ventilators of these aerotherms are controlled by thermostat. This equipment assures that the compartments will warm up very quickly and makes it unnecessary to use any separate source of heat.

The above are the principal characteristics of the three types of railcars retained by the French National Railways to complete and renew their stock of railcars now in service; as we mentioned above they will be completed by a certain number of small capacity country vehicles intended especially for certain small lines which remain open to passenger traffic.

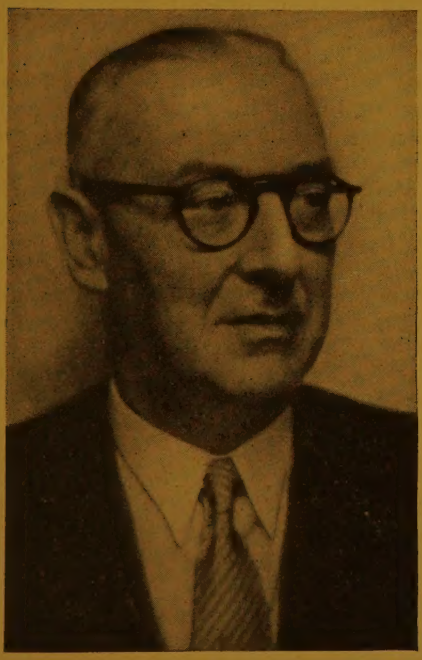
The construction of the above mentioned railcars, hindered by the shortage of materials, has only just begun; it is to be hoped that these new units will soon be put into service to reinforce the present stock which, in spite of certain imperfections and the damage sustained, has played a very important part in restoring the passenger services since the liberation of the country.

[385. (09.2]

OBITUARY.

Justin COLLE,

Operating Manager of the Belgian National Railway Company.
Member of the Permanent Commission of the International Railway Congress Association.



It is with great regret that we have to report the untimely death on the 16th June of M. Justin COLLE, Operating Manager of the Belgian National Railways, President of the IVth Commission of the International Railway Union, and member of the Permanent Commission of our Association.

M. Justin COLLE was born in 1892. After a brilliant scholastic career, he obtained his diploma as civil mining engineer at Liége University, and electrical engineer at the Montefiore Institute.

He entered the service of the Belgian State Railways in 1919 as an engineer in the Rolling Stock Department, where his brilliant qualities soon attracted the attention of his superiors.

Transferred to the Operating Department in 1927, he gained very rapid promotion and finally became manager.

He was able to use his remarkable organizing faculties to the utmost in the restoration of the railway after the liberation of Belgium in 1944. The important part he played in the organisation of military transport and the exceptional services he rendered in this connection earned him the most flattering testimonials from the Allied Governments.

Well known and much appreciated in international railway organisations, he was a member of the Permanent Commission of the International Railway Congress Association, and a member of the International Timetables Conference on which he was president of the Nord-Express and Ostend-Vienna Express groups. He also represented the Belgian National Railways in the RIC and RIV Unions, as well as on other international organisations.

In 1946, he was chosen to preside over the IVth Commission of the Inter-

national Railway Union. His great personal charm, the soundness of his judgment, his broadmindedness and courtesy, and the tact he employed in all these official capacities justly won him the highest estime and sympathy of all the delegations.

M. COLLE was elected a member of the Permanent Commission of our Association in 1946 and always took the greatest interest in its work. His participation in the work of the Railway Congress dates back to the Paris Session in 1937 where he was the Reporter for Question VIII « Application of rational methods of organisation (planning) to goods transport ». We would also recall the important part he played during the discussions on the important question of the organisation of passenger train services at the recent Lucerne Session (1947).

His remarkable career was recognised by the award of many Belgian and foreign orders : he was Officer of the Order of Leopold, and also held the « Medal of Freedom » and the British Empire Cross, as well as other decorations.

An eminent official and a striking personality, very upright, full of energy and of a rare degree of competence, his loss will be felt both in Belgium and abroad by all those who had the privilege of knowing and working with him.

We offer his family our sincerest condolences.

The Executive Committee.

NEW BOOKS AND PUBLICATIONS.

[621 .13 (02]

LAMALLE (Ulysse), Ingénieur Civil des Mines, A. I. Lg., Directeur Général honoraire de la S. N. C. B., Professeur du cours d'Exploitation des Chemins de fer à l'Université de Louvain, Membre d'honneur de l'Association Internationale du Congrès des Chemins de fer, and LEGEIN (Fernand), Ingénieur Civil des Mines A. I. Br., Ingénieur en Chef honoraire de la S. N. C. B. — **La Locomotive. Description raisonnée de ses organes** *(The Locomotive. A methodical description of its component parts)* (4th edition). — One volume ($7 \times 9\frac{1}{2}$ inches) of 770 pages, with 822 figures. — 1948, Louvain, Librairie Universitaire Charles Uystpruyst, 10, rue de la Monnaie, et Paris, Dunod, 92, rue Bonaparte. (Price : 400 Belgian francs.)

When this book first appeared in 1913, it was enthousiastically welcomed by the traction staff of the Belgian railways. Those who, like the writer of these lines, have had frequent discussions with the shop staff and train staff have often had occasion to note how widely this book has been read and appreciated. It is invoked, it is consulted, it is used to prepare for examinations for promotion.

There is no need to be astonished at this. At that time, and still to-day the work of Messrs. LAMALLE and LEGEIN is the only reference book available on this subject in Belgium. To say that it fills a gap is an understatement; it does away with a complete void. The reason for its ever growing fame and popularity are to be found in the intrinsic merits of the book itself. The authors who are primarily traction men in spite of their other activities have a full knowledge of their readers and of their material. They know the needs of those who come in contact with the actual facts and have had no special preparation. Their faculty of analysis and their talent for explaining have done the rest.

To be true certain descriptions and explanations can be found in the administrative instructions which are very clearly worded and made clearer by drawings and the necessary diagrams, or in the notices issued by the suppliers of various equipment. But these are scattered documents, and whatever their value in themselves, they can only be fully appreciated by those who can collect together a methodical documentation. For the others the real instrument of study is the book.

The book now being reviewed has the inestimable advantage for readers who are not very well informed of giving them a choice made by enlightened writers, a choice covering not only the subject dealt with but also the ideas developed and the way they are presented. A whole series of fundamental data, both theoretical and practical, covering various fields, such as physics, general mechanics, mechanics of heat, knowledge of materials, makes it unnecessary for the reader to look for further explanations elsewhere. They make it possible for him to understand the arrangements adopted and the working of the numerous parts of an engine which is at one and same time a generator of steam, a steam engine, and a vehicle running on rails.

No doubt the other side of the question is the stability, the finality of a book whereas technique evolves, often very rapidly. Here the true worth of the authors can be appreciated for they have profited by the chance of each

new edition to rejuvenate their work by making the modifications and additions experience has shown to be necessary.

Twenty years have passed since the *Bulletin* reported the publication of the third edition, twenty years in which many alterations and innovations have appeared. It is M. LAMALLE, since the loss of his co-author M. LEGEIN, who has prepared the fourth edition of the book.

Owing to force of circumstances the traction stock has become very varied. New devices have been introduced, some of them inspired by American practice. Steam distribution has been the subject of much research work leading to the use of valves. Lubrication, a matter of great interest to the engineman, has been perfected. Attempts have been made to facilitate servicing and to improve cleaning and maintenance. There is a greater diversity of special braking equipment on different types of locomotives. Superheaters and feed water heaters of various kinds have been used. The question of running through curves has become of greater interest owing to the higher speeds. And so on. This partial summary explains why the book is so much larger. Those interested will find therein the means of making themselves familiar with all the old or new devices they may come across.

It is certain that the fourth edition will be just as successful as its predecessors.

E. M.